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REPAIRING GERMAN VANDALISM BY ELECTRIC WELDING.

BY COMMANDER E. P. JESSOP, U. S. NAVY, MEMBER.

On the 6th of April, 1917, the interned German and Austrian vessels located in the port of New York were taken over by the collector of customs for safe keeping, the crews being sent to Ellis Island and interned. These vessels consisted of the *Vaterland*, *Kaiser Wilhelm II*, *Koenig Wilhelm II*, *President Lincoln*, *President Grant*, *Barbarossa*, *Grosser Kurfurst*, *Hamburg*, *Harburg*, *Pisa*, *Nassovia*, *Frederic der Grosse*, *Prinzess Irene*, *Adamsturm*, *George Washington*, *Martha Washington*, *Magdeburg*, *Prinz Joachim*, *Allemenia*, *Prinz Eitel Frederich*, *Pennsylvania*, *Armenia*, *Portonia*, *Maia*, *Clara Mennig*, *Dora*, *Ida*, *Himalaya*, *Indra*, *Mattador*, *Bohemia*. These vessels were located at various places in the port, 5 being at 135th Street, 15 of them at the Hoboken piers, 2 at 29th Street, Brooklyn, N. Y., 4 at 65th Street, Brooklyn, N. Y. *Martha Washington* at Stapelton, Staten Island, *Himalaya* at Newark Bay. Of these the *Martha Washington* and *Him-*

alaya were in the courts under libel and therefore were not available for our use.

Immediately on taking over these vessels the Shipping Board appointed a committee of engineers to make a survey of the vessels and superintend the repair of them, while the Collector's office was responsible for their care and preservation until such time as they were taken over by the Shipping Board for repairs.

The organization of the Collector's office did not have the necessary technical experts to care for these vessels, and an officer was therefore detailed by the Commandant, New York Navy Yard, to assist the Collector in this respect, and he acted in that capacity until all the vessels were turned over to the Shipping Board.

It had been announced in the public press previously that the machinery of these vessels was very badly damaged and an inspection of these damages was made by a board of naval officers appointed by the Commandant shortly after the damage was discovered and some time before this country entered the war.

This inspection developed the fact that the vandalism on all vessels was, in general, the same, which would indicate that general instructions had been issued to all commanding officers by the same agent. It also appears that the parts damaged, as a rule, were those probably on hand as spare parts in the home ports of the vessels. In case this country did not enter the war, the ships could be again commissioned by utilizing spare parts already provided.

It was appreciated that the rapid repair of these vessels was of the greatest importance and to that end the Collector appointed a Board of Railway Engineers to investigate the damage to the machinery to these vessels with a view to the use of electric welding where it could be done satisfactorily. This Board, headed by the Consulting Engineer of the Erie R. R., consisted of the Electrical Engineer of the New York Central, the Mechanical Superintendent of the Erie, one general foreman from the Erie, and one general foreman from the New York

Central. These gentlemen were all thoroughly familiar with the use of electric welding on the railroads. The railroad engineers were probably the only ones in this country who had made any extensive successful use of electric welding in connection with cast iron. Their services were offered to and accepted by the Collector.

The Shipping Board had also appointed a Board of Engineers to inspect these vessels. The inspections by these two Boards were made at the same time, although the Boards worked separately. The report of the Board of Engineers for the Shipping Board recommended the renewal of all cylinders which were badly damaged. These recommendations were made without any idea of the use of electrical or other methods of welding and were therefore not of any use as to time and cost if welding processes were applied. The report of the electric welders committee was unanimous in stating that all damaged cylinders could be reclaimed. This report was made after this committee had been cautioned to be very conservative in their statements with regard to what they could do, as it was important that the work be completed in such manner as to render the engines of these vessels as reliable after repair as they had been originally. The electric welders committee reiterated their report that all the cylinders could be saved.

Great opposition to the electric welding of cast-iron cylinders developed among certain engineers and it was with the greatest difficulty that consent was obtained for a trial of this method. With so much at stake, and considering the time element, the logical procedure would be to repair one cylinder by this method and to put this cylinder through a series of drastic tests. Unfortunately this methed was not pursued, and the controversy with regard to the methods to be applied in these repairs continued until the larger vessels were turned over to the Navy Department to be fitted as transports, about the middle of July, 1917.

All told, the principle of electric welding has been applied to 15 ships in the port of New York. Of these 15 ships all

are in commission and will probably be ready for service before this article appears in print. In no case known to the writer will the repairs to the propulsive machinery of the vessels delay them beyond the time necessary to equip them as transports. In other words, by the use of electric welding all of the vessels will have been gotten out as quickly as would have been the case had no damage to the main machinery been done by the Germans, the vessels being in other respect badly run down and in need of extensive overhaul.

At first glance it would appear that the skepticism of marine engineers toward the electric welding of iron castings would indicate a lack of progressive ideas in the marine world, when it is a fact that the railroads have been using this method successfully for several years. This, however, is not necessarily the case, inasmuch as marine repairs do not bring into play the repair of cast iron as frequently as do railroad repairs. Cracking of cylinders is one of the most common of locomotive breakdowns. This, added to the imperative need of some method of quick repair for railroad equipment, has served to develop in this particular branch of engineering the use of electric welding far beyond its development in the marine world. It would appear that the engineering magazines have not kept pace with the times, or the great use which railroads are making of electric welding would have been more exploited and the knowledge of this particular method of repair more generally disseminated. It is also true that the mechanism used in electric welding has had much to do with propagating the idea that cast iron could not be welded. The ordinary welding apparatus has not sufficient control of the electric current to insure proper heat at the point of the weld, thereby rendering electric welds very unreliable. For this reason the railroads were forced to expend large sums of money in developing a welding apparatus which did have this control, and they have today what seems to be a satisfactory equipment from this viewpoint.

In order to successfully weld with the electric arc it is necessary to have complete control of the current. The arc

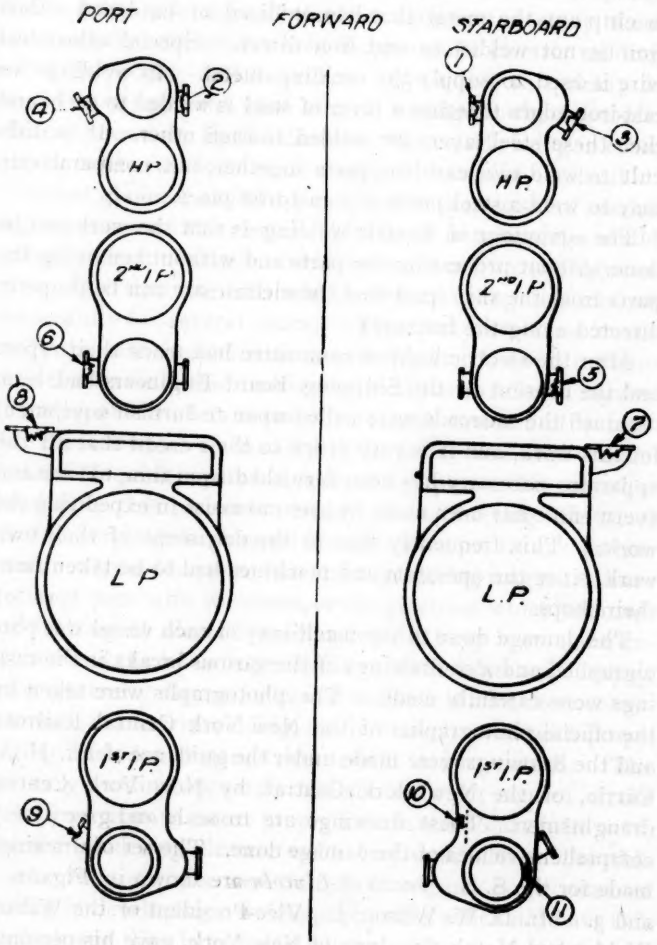
must be directed along the line of fracture to be welded. Welding must be done slowly, the metal being laid on layer by layer, and each layer must be caulked and peened to knock or chip out the metal that has oxidized or hardened. Cast iron is not welded to cast iron direct. Special alloy steel wire is used to supply the welding metal. In welding two cast-iron edges together a layer of steel is welded to each, and then these steel layers are welded to each other. It is difficult to weld two cast-iron parts together, but comparatively easy to weld a steel piece to a cast-iron piece.

The advantage of electric welding is that the work can be done without preheating the parts and without removing the parts from the ship (provided the electric arc can be properly directed along the fracture).

After the electric welders committee had made their report and the consent of the Shipping Board Engineers had been obtained the railroads were called upon to furnish equipment for this work, and it is very much to their credit that all the apparatus necessary has been furnished from time to time and every effort has been made by them to assist in expediting the work. This frequently was to the detriment of their own work, since the operators and machines had to be taken from their shops.

The damage done to the machinery of each vessel was photographed and also drawings of the various breaks in the castings were carefully made. The photographs were taken by the official photographer of the New York Central Railroad and the drawings were made under the guidance of Mr. H. A. Currie, of the New York Central, by New York Central draughtsmen. These drawings are to scale and give a very comprehensive idea of the damage done. The set of drawings made for the S. S. *President Lincoln* are shown in Figs. 1, 2 and 3. Mr. D. W. Wilson, Jr., Vice-President of the Wilson Welder and Metals Co., Inc., of New York, gave his personal attention and supervision to the work of welding.

The first vessel upon which this method of repair was tried was the *Armenia*, which had been taken to a private ship


PORT
FORWARD
STARBOARD
H.P.
2ND I.P.
L.P.
1ST I.P.
H.P.
2ND I.P.
L.P.
1ST I.P.
1
2
3
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5
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11

FIG. 1.

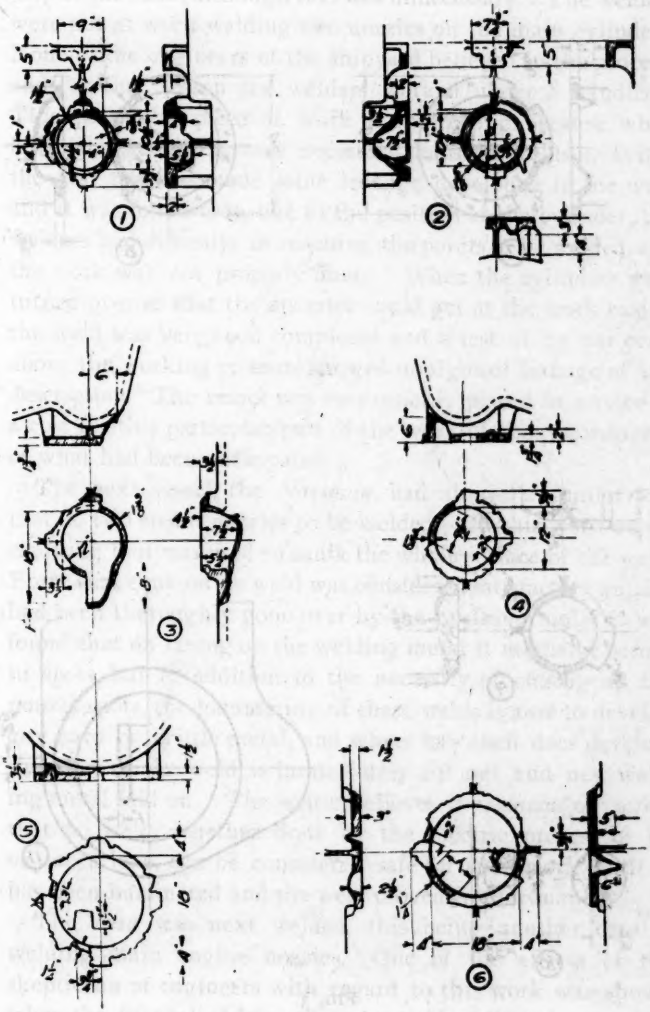


FIG. 2.

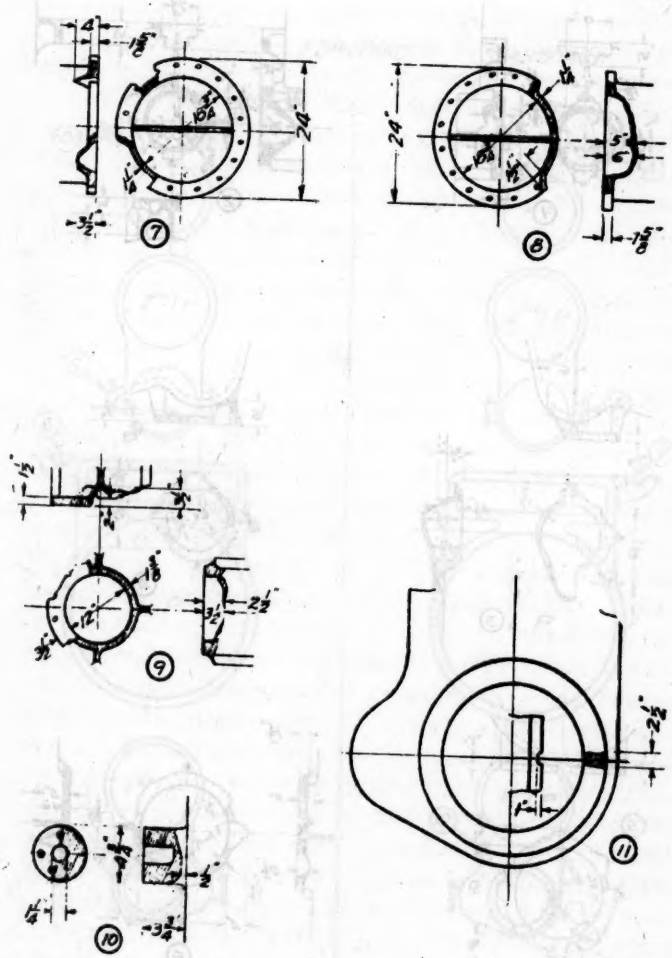


FIG. 3.

yard for overhaul. The cylinders were removed from the ship to the dock, although this was unnecessary. The welders were put at work welding two nozzles on the main cylinders. None of the engineers of the shipyard believed in the process, and for that reason the welders worked under a handicap. This particular piece of work is interesting because when finished no caulking was necessary on these welds. When the first test was made some leakage developed in the weld and it was found that, due to the position of the cylinder, the welders had difficulty in reaching the points to be welded, and the work was not properly done. When the cylinders were turned over so that the operator could get at the work easily, the weld was very soon completed and a test of 75 per cent. above the working pressure showed no signs of leakage of any description. The vessel was very quickly placed in service at a cost for this particular part of the work of about one-fourth of what had been anticipated.

The next vessel, the *Nassovia*, had almost a similar job, that is, two steam nozzles to be welded. On this work an air caulking tool was used to caulk the whole surface of the weld. From that time on no weld was considered satisfactory until it had been thoroughly gone over by the caulking tool. It was found that on laying on the welding metal it might be porous in spots, but in addition to the necessity of closing up the porous spots, the hammering of these welds is sure to develop any hard or brittle metal, and where any such does develop, that part of the weld is immediately cut out and new welding metal laid on. The writer believes it *essential for safety* that no weld, whether done by the electric process or by oxy-acetylene, can be considered safe for such work until it has been hammered and the weld proved in that manner.

The *Pisa* was next welded, this being another case of welding main engine nozzles. One of the effects of the skepticism of engineers with regard to this work was shown when the *Pisa* sailed from this port to Norfolk and, on arrival at Norfolk, it was reported that the weld had given way. Investigation demonstrated that two small pin holes had

shown up in the weld from which oozed an occasional drop of water, no steam issuing whatever, but there had been so much talk of the dangers of the electric welding that the engineers of the vessel were afraid to go to sea with it. It was only after the naval engineers from the Norfolk Navy Yard had pronounced the work safe that the engineers of the ship were prevailed upon to make the voyage.

The *Prinzess Irene* and the *Frederic der Grosse* were sent to the Navy Yard, Brooklyn, N. Y., for overhauling. Complete photographic records of the work done on the main engines of these vessels were kept, and some are embodied in this article. The breaks shown on these engines were probably as difficult of repair as any. These two vessels have been completed and are now in the transport service. From the start no trouble has ever developed with regard to the welds after they were pronounced finished. Some of these photographs show the break after being prepared for welding; others show the cylinder with the insert fitted ready for welding, and others show the completed weld. It will be noted that the cast-iron side is studded. This is not done as a necessary part of this repair, as all tests of welds made by the welding apparatus have shown practically the same tensile strength as the material which was welded. This laying over of metal on the cast-iron side of the weld was merely done as a precaution and to increase the strength of the weld beyond that of the rest of the cylinder. Studding up the cast-iron side of the break has also been the subject of much criticism, numerous engineers having jumped to the conclusion that the repair was not made by welding the cast-iron nearly so much as it was made by welding around steel studs which had been embedded in the cast iron. This, however, is not a criticism which is vital, because it was only necessary that the cylinder welded be rendered as strong as before the break, and this undoubtedly has been accomplished in every case.

The welding done on the *Frederic der Grosse* and the *Prinzess Irene* is the first straight electric welding in any of these vessels. In all previous cases the skepticism of the

FIG. 4.--PRINZESS IRENE, DAMAGED H. P. CYLINDER, READY FOR INSERT, PREPARATORY TO WELDING.

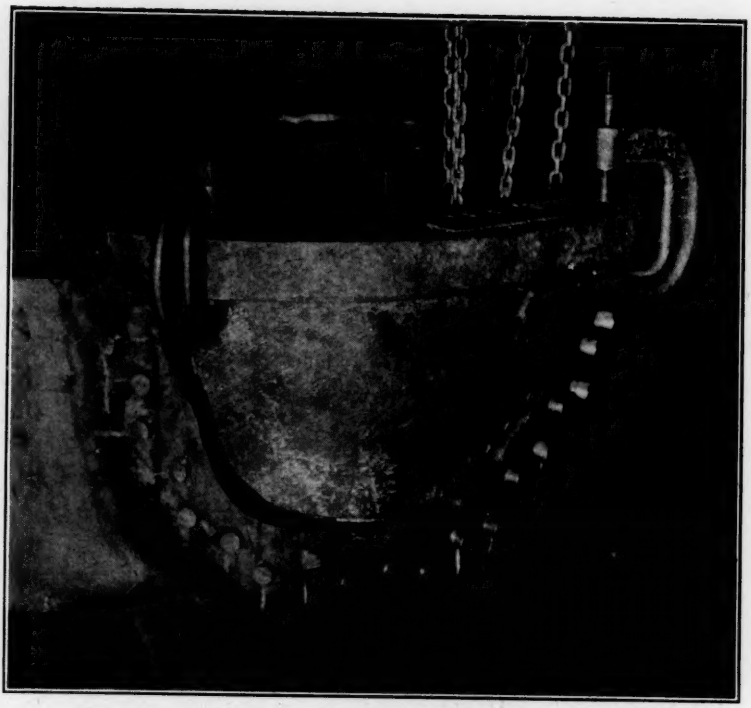


FIG. 5.—SAME AS FIG. 4, INSERT IN PLACE READY TO WELD.

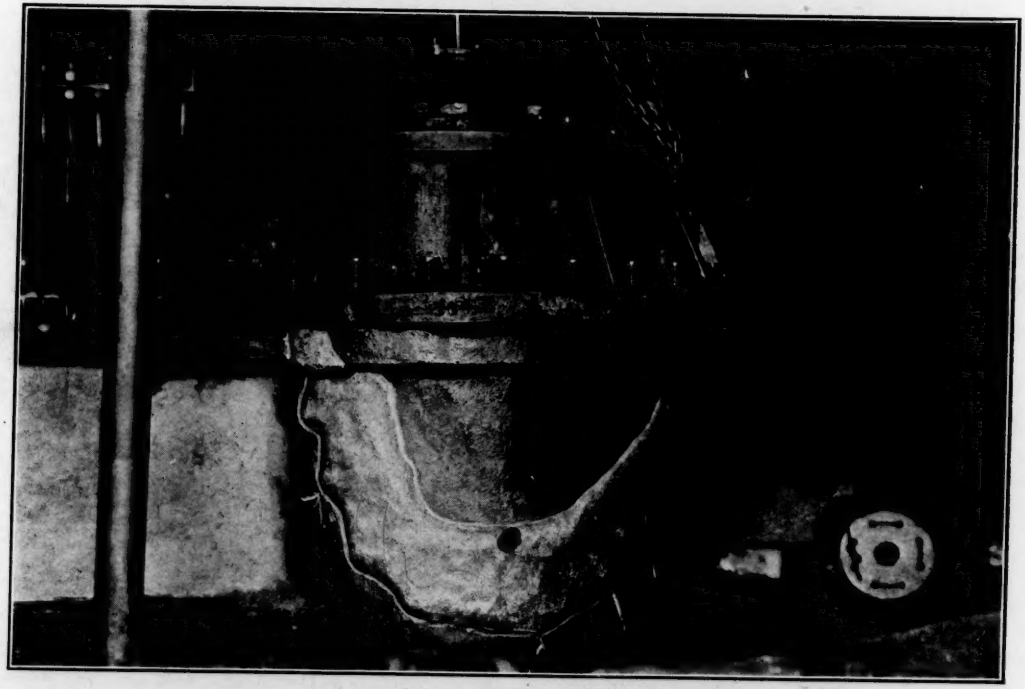


FIG. 6.—SAME AS FIG. 4, FINISHED WELD.

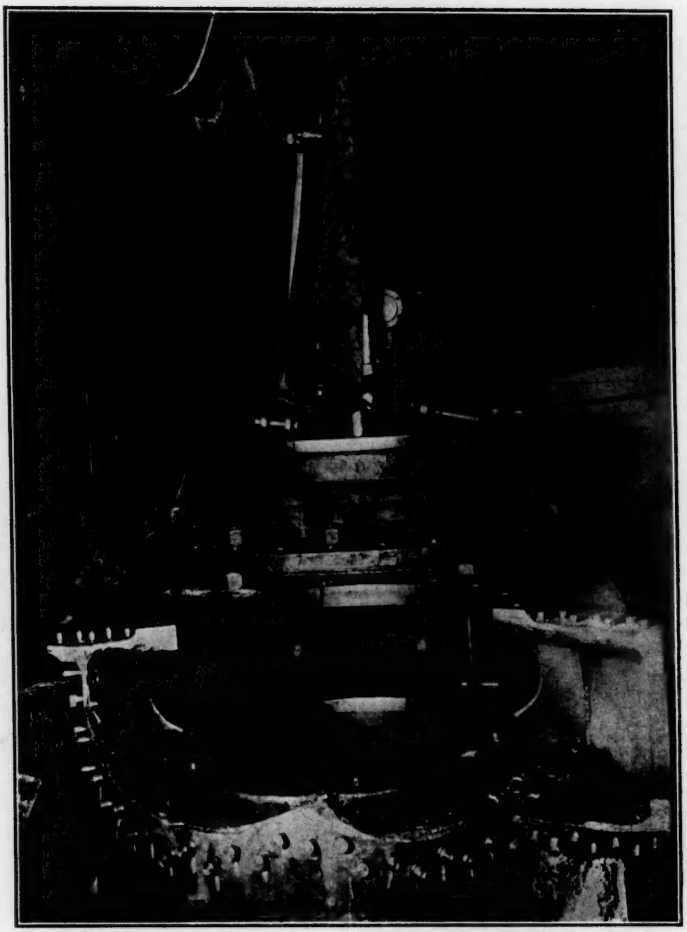


FIG. 7.—PRINZESS IRENE, DAMAGED I. P. VALVE CHEST, READY FOR INSERT, PREPARATORY TO WELDING.

FIG. 8.—SAME AS FIG. 7, INSERT IN PLACE, READY TO WELD.

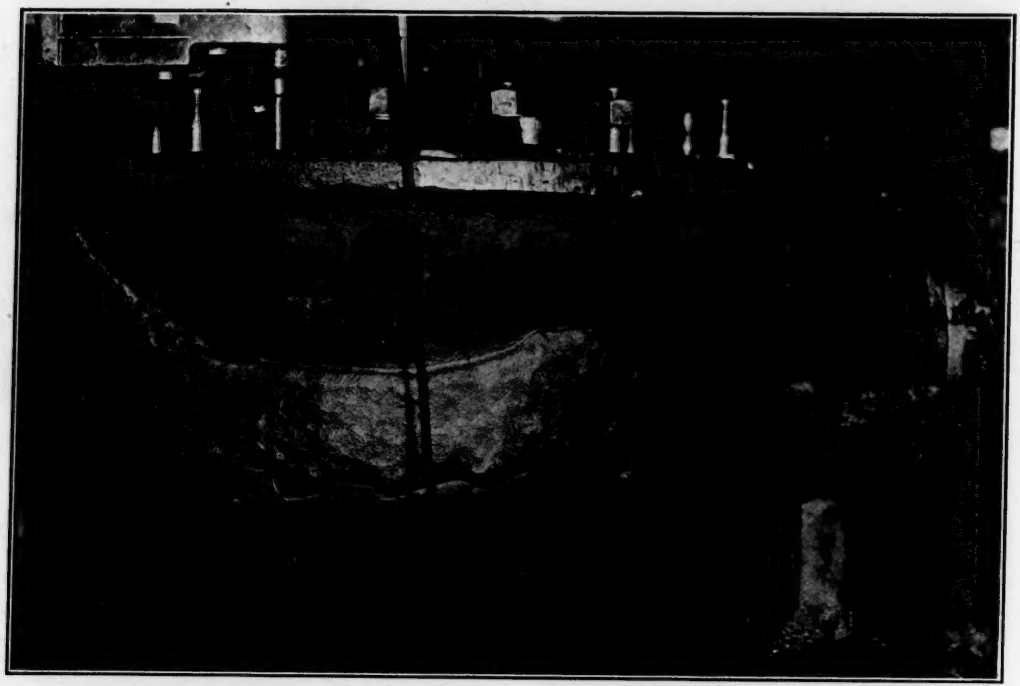


FIG. 9.—SAME AS FIG. 7, FINISHED WELD.

engineers who had immediate control of deciding upon the form of patches which should be used caused them to have patches cast overlapping the cylinder walls and there bolted to the walls, so that part of the pressure was taken by the bolts and not all of it by the weld, as is the case of the *Frederic der Grosse* and the *Prinzess Irene.*

On the *Koenig Wilhelm II* the original intention was to repair the cylinders by mechanical patches. These patches were made of composition and in most cases were in the form of inserts between cast-iron sides of the breaks. When these vessels were taken over by the Navy Department all composition patches were discarded and cast-steel patches substituted, but, due to the fact that the cylinders had already been tapped for the bolts of these mechanical patches, the cast-steel patches were bolted to the cylinders and then welded. The amount of fitting necessary to make a mechanical patch steam tight is shown by the fact ten weeks after the *Koenig Wilhelm II* repairs had started the machinists were still at work on these patches scraping them to a fit. The difference between this method and that of welding steel patches is shown by the fact that many of the patches on the *Frederic der Grosse* and the *Prinzess Irene* were fitted in 48 hours ready for welding. In other words, they are not fitted at all, but merely bevelled off to receive the welding material.

On the *Kaiser Wilhelm II* a slightly different condition was encountered in that the liners for the cylinders in some cases had pieces broken out of them. In the case of the low-pressure liners it was decided that the small area covered by the break did not justify the condemning of these liners. Therefore cast-iron inserts were made and welded in place and then ground down to the surface of the rest of the liner. New liners were cast for the high-pressure cylinders.

Besides the damage to the cylinders and valve chests there were some cases of bent piston rods, broken pistons, broken cylinder covers, and various other smaller items of destruction. On the *Kaiser Wilhelm II* the intermediate piston rod on each engine was badly bent, but not so badly as

to preclude its being straightened and replaced. This work is now completed. In no cases were the boilers of these vessels maliciously injured, and in very few cases were the major auxiliaries tampered with. From the methods used in disabling the machinery it would appear that the Germans had a definite plan in view, and it is very probable that all parts broken have duplicates already cast and ready in Germany to replace them at the end of the war. In a number of the ships the engines were disabled by breaking or throwing overboard the throttle valve. The procuring of new throttle valves has generally taken as much time as the repairs of the breaks to the machinery.

The case of the *Vaterland* is quite different from that of any other of the ships. The *Vaterland*, being the largest ship afloat and the most complicated, seems to have had inferior engineering talent operating it. Her engines are of the Parsons type of turbine, there being four ahead and four astern turbines on four shafts. All the ahead engines were found in excellent condition and all of the astern engines were found more or less damaged. The major part of this damage seems to have been due to faulty operation and not to any malicious intent. Cracks were found in the casing of the starboard H.P. backing turbine. These were so extensive that it was palpable that the engines had not been used on the last run. This view has been corroborated since by the discovery of certain written matter on board showing that the vessel made the last trip on three propellers and at reduced speed. One crack in the lower casing of this turbine extended a distance of about 8 feet around the circumference of the steam belt. Other cracks were found in the upper half of the casing. In addition to this the dummy cylinder was found cracked in three places, all being in the wake of the nozzle box. This turbine has one stage of impulse blading, and, as the action of the nozzle boxes had evidently been such as to distort the casing and develop the cracks which were fouud, it was decided to cut off the impulse blading entirely. While this will reduce the economy in these engines, it is not believed

that it will reduce the backing power to any extent, since more steam will be admitted to the reaction blades than formerly.

The auxiliaries of the *Vaterland* were in fair condition, as were the boilers. The vessel has 46 Yarrow boilers with closed ash-pit system of forced draft. The boilers showed evidence of poor handling. The presence of a great amount of oil shows poor marine practice, and it appears that the Germans were using lubricant in the cylinders of the auxiliaries. This practice has been discontinued in our service for many years. It is a curions fact that, although this vessel had steam lines which in some cases extend over a length of 300 feet, she was not fitted with proper drains and traps, such drains as were fitted being run into the bilges. The boilers are not fitted with internal feed pipes, and as a result they seem to prime very easily. Internal feed pipes have been fitted. Beside the damage to the blading of the backing turbines a few minor attempts at disabling the ship were discovered. One section of suction pipe to the fire and bilge pumps was found with holes bored in it, these holes plugged up with putty and red lead and then covered so that they would not be discovered. The glands for the stern-tube stuffing box had been removed and strewn through the bilges. A great deal of water had entered the stern-tube compartment. All parts of these glands were located and replaced and the compartments cleared of water.

One of the great difficulties in carrying on the repair work of these vessels has been the lack of proper drawings and the lack of information with regard to the operation of the machinery. This has been particularly true in the case of the *Vaterland*, on account of the great mass of machinery and electrical apparatus and piping, and it has been only by the most patient and dogged following up of pipe lines and electric wiring that sufficient knowledge of this vessel has been acquired to render it safe to take it to sea. All of her electrical equipment is of the very latest type, some of it manifestly superior to that in use in our service. One item which

the writer noticed particularly is the micrometer attachment to the gyroscopic compass by which the helmsman can note the change of one-tenth of a degree in the heading of the ship. I am informed by the gyroscopic experts of the New York Navy Yard that the German compass is very much more simple and needs very much less adjustment and care than the one in use in our service.

The complicatedness of the electrical system of the ship can be judged when it is stated that there is a clock system connected electrically with the master clock on the bridge which controls and regulates 530 clocks throughout the ship. The vessel also has a complete watertight door system, the doors being operated hydraulically and controlled from a station on the bridge. For fire protection all bunker compartments below are connected by tubes to a glass case on the bridge. A system of blowers, operated at regular intervals, discharge air through these tubes into an indicator box. In case there is fire in any compartment, smoke is carried by this blower into the indicator box on the bridge showing what compartment is on fire. Alongside of this box is a hose attachment for attaching to the pipe from the compartment, and valves for turning on a smothering gas to extinguish the fire.

This vessel is not a cargo carrier and therefore any coal which is used must be compensated for by water ballast, so it has the necessary tanks for keeping on an even keel and also has a set of anti-rolling tanks for use in a sea-way to prevent excessive rolling. These anti-rolling tanks were fitted on account of the instability of the *Imperator* at sea.

In summing up the subject it is thought necessary to invite particular attention to the following items with regard to electric welding and its proper application. In welding cast iron it is particularly essential that the welding metal be laid on the cast-iron surface with the greatest care, since it is the juncture between the cast iron and the welding metal which is liable to be inferior. To accomplish this the welding metal should be laid over the cast-iron surface wherever possible before the patches are put in place. This method gives the

welder more room to work and permits him to keep a steadier arc and thus reduce the probability of too much or too little heat spoiling that part of the weld. This metal as it is laid on the cast-iron surface must then be peened by a caulking tool in order to develop any hard spots or spots where contact is not good. After this part of the operation is completed the patches should be put in place, and the completion of the weld then is merely welding of steel to steel, which is a very simple process.

Of course it is important in such special work as this that the operators be the most skillful obtainable. In the work which has been carried on in the German ships the operators have not been particularly skilled men in all cases on account of the great number necessary and the comparatively few who are fairly skilled in this art. For that reason more cutting out and replacing of bad metal has been necessary than would have been the case if the work had not been so extensive and had only skilled operators been used.

It is thought to be a remarkable fact that in all the work which has been undertaken by the electric welders there has never been a failure, this in face of the fact that almost every conceivable kind of patch has been handled, and also in face of the fact that the work has been very much more extensive than any ever before attempted. I believe this is practically the first of its kind of any moment to be attempted in marine engineering. The writer's confirmed belief is that the scraping of cast-iron parts of machinery is entirely unnecessary and will very soon be a thing of the past. It cannot be too strongly insisted that in every case in which this method has been used the repaired job has been fully as strong as it was before the damage was committed, and in most cases stronger in the particular part affected.

BOILER DAMAGE ON S. S. *POMMERN.*

The S. S. *Pommern*, now the *Rappahannock*, was interned at San Francisco. When it was decided to put her in service a survey of the vessel developed the fact that of four Scotch boilers on the vessel but one could be repaired. It was necessary to renew the other three.

The accompanying photographs give some idea of the nature of the damage. The boilers were ruined by dry firing. Not satisfied with the results of this treatment, it appears that thermit or some kindred agent was employed to complete the vandalism.

The net result, however, was negligible. Considering, quite aside from the cash value, the military value of the vessel, for every ship today has at least a potential military value, the cost and time required to effect the necessary boiler repairs and replacement would surprise the agents that so carefully thought out the campaign of vandalism on the German vessels seized in this country.—EDITOR.

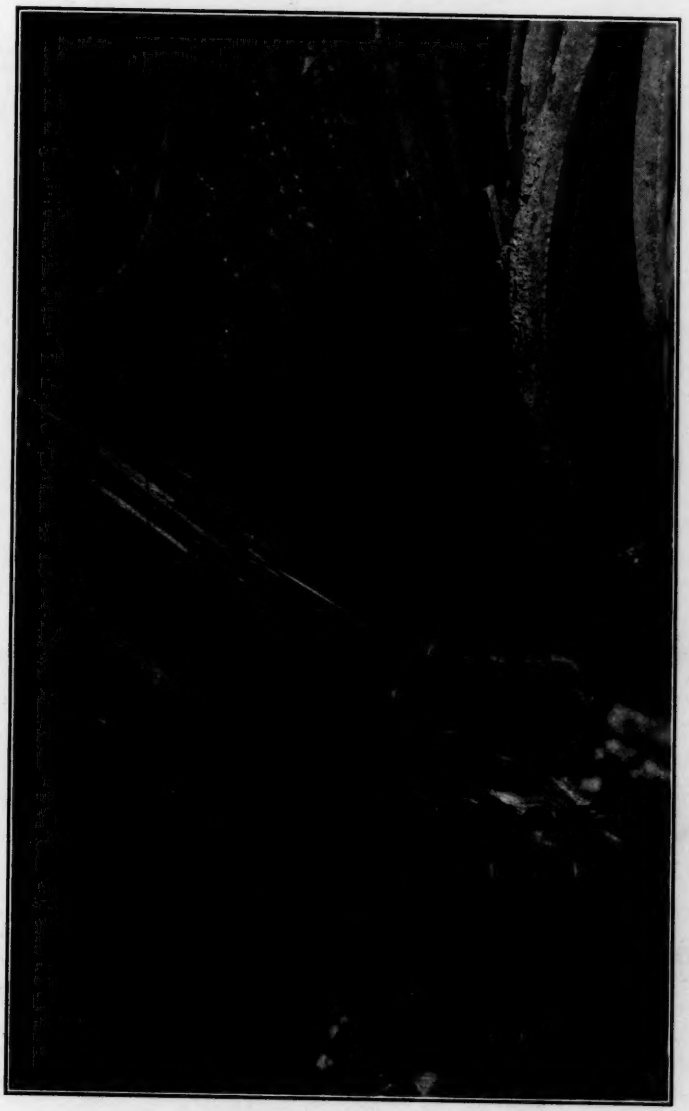


FIG. 1.—DAMAGE TO "POMMERN'S" BOILERS.

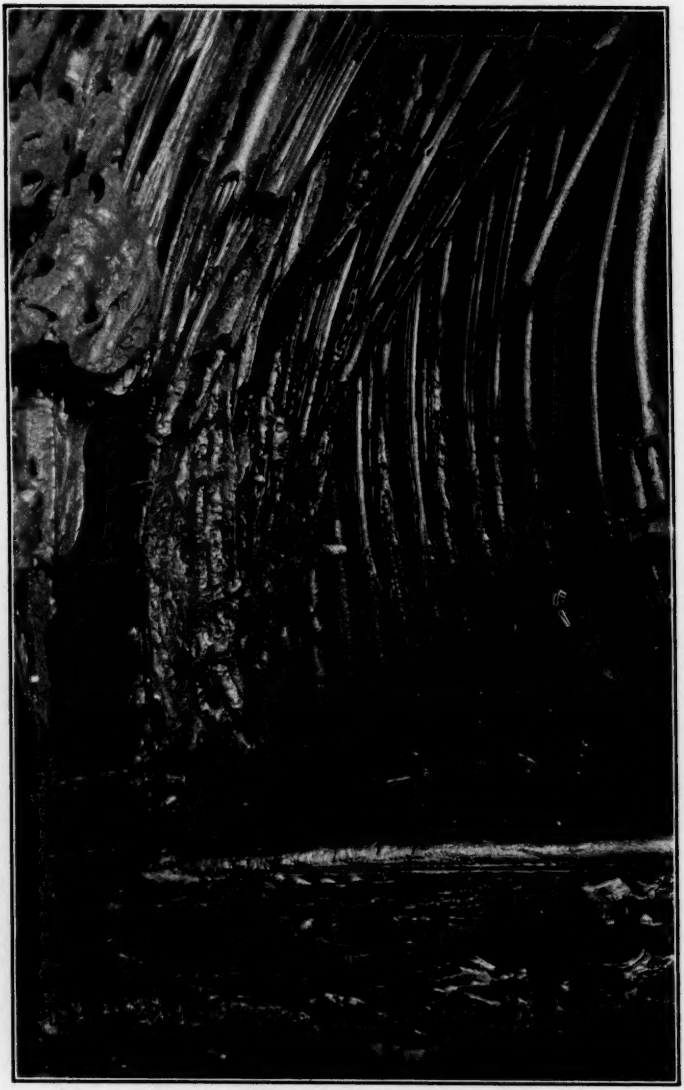


FIG. 2.—DAMAGE TO THE "POMMERN'S" BOILERS.

WELDING MAIN ENGINE CYLINDERS.

GERMAN SHIPS "NECKAR," "RHEIN" AND "BULGARIA."

By Naval Constructor H. G. Knox, U. S. Navy.

When the *Neckar, Rhein* and *Bulgaria* were turned over to the Norfolk Navy Yard for repairs their main engine cylinders had been badly damaged by the German crews just prior to seizure of the ships by the United States Government. The ships are all twin screw and equipped with quadruple-expansion engines of the usual marine type. Of the 24 cylinders on the three ships, 17 were damaged. Of this number 15 required welding. This applies to the cylinders proper. In addition there were several broken fittings and cracked cylinder liners. In the course of repairs the only casting which was made complete was one low-pressure valve-chest cover, which was so badly broken as not to warrant repair.

On the *Bulgaria* the two high-pressure cylinders were damaged by having the flange to the main steam inlet nozzle broken. These two cylinders were repaired by machining off the broken flanges and threading the nozzles with a fine thread for about 2 inches. Steel-plate flanges were turned, threaded and fitted to the nozzles. No welding was performed on these two cylinders, and the job tested satisfactorily.

A brief description of the welding effected on the other cylinders is itemized as follows:

"NECKAR" (U. S. S. "ANTIGONE").

Port high.—A cast-steel patch about 48 inches long, including the cylinder wall and top flange of the cylinder, was first arc-welded in place. The cylinder was then finished up and tested to 200 pounds hydrostatic pressure. The patch leaked

to such an extent that even with caulking it was impossible to make the patch tight. This patch was then cut out below the weld and a cast-iron patch acetylene-welded in place. This patch was about 60 inches long and developed no leaks when tested.

Starboard high.—A 32-inch cast-iron patch was acetylene-welded in this cylinder and a section of flange 39 inches long was also welded in place.

Port first intermediate.—A narrow cast-iron patch, including the top cylinder flange, was acetylene-welded in this cylinder.

Starboard first intermediate.—A cast-iron patch 51 inches long, including the top flange of the cylinder, was acetylene-welded in this cylinder.

Port second intermediate.—A 54-inch break was repaired by a cast-iron patch in the top cylinder wall. A 36-inch section of flange was also acetylene-welded in place. A piece broken out of the cylinder casting in the slide-valve space was electric-welded. This break was about 24 inches long.

Starboard second intermediate.—A large section of cylinder wall was broken out of this cylinder. The seam in the cast-iron patch was 102 inches long, acetylene-welded. One pin hole developed on test and was plugged. No other leaks developed. Two 16-inch sections of flange in rear of lugs were also welded in place. One 9-inch piece was broken out of slide-valve seat and acetylene-welded.

Port low.—Two broken pieces of cast iron, of about 28 inches each, were electric-welded on the top end of the cylinder over steam port. One section broken out of cylinder casting next to slide valve was also electric-welded. This patch was about 24 inches long.

Starboard low.—The starboard low-pressure cylinder was broken in a similar manner to the port. Two pieces, each 28 inches in length of weld, were welded in top end of steam port, and a 24-inch section in the valve-chest wall. All of these welds were made by electric arc.

Steam line elbows.—Two sections of flange, 9 inches long each, were acetylene-welded in cast-iron high-pressure, steam-line elbows. The broken pieces were rewelded in place.

"RHEIN" (U. S. S. "SUSQUEHANNA").

Port high.—A comparatively small patch was welded at the upper end of this cylinder, and included a section of flange. An axial crack extending 6 or 8 inches from the crack was also included in the weld. The total length of the weld was about 28 inches, and was made by acetylene.

Starboard high.—Undamaged.

Port first intermediate.—A 38-inch cast-iron patch was acetylene-welded, and included a section of flange. One small pin hole developed on test and a screw plug was inserted.

Starboard first intermediate.—Undamaged.

Port second intermediate.—A 46-inch cast-iron patch, including a lug, was cast in iron and welded in place by acetylene. The liner of this cylinder was broken in pulling out of cylinder and an 8-inch length of patch welded in top edge of liner. The only leaks developing on test were a pin hole in the weld and a leak in the casting outside of weld where the lug was not sufficiently filletted. Both leaks were easily closed without welding.

Starboard second intermediate.—A large cast-iron patch, including a lug, was acetylene-welded as shown in photograph. The main break was 73 inches long and in addition two 13-inch sections of flange and an 8-inch section of cylinder liner at top similar to the port second intermediate were built up.

Port low.—A large cast-iron patch, including a section of top cylinder flange, was acetylene-welded in this cylinder. The length of seam was about 80 inches, and the weld was made by acetylene. The liner for this cylinder had an L-shaped crack extending from the top down and to the left, total length of crack, about 27 inches. This was welded by acety-

lene, as were also two 8-inch pieces broken out from the top edge of liner.

Starboard low.—A 73-inch cast-iron patch was acetylene-welded in this cylinder and a 19½-inch axial crack was acetylene-welded in liner.

"BULGARIA."

In addition to the two high-pressure cylinder nozzles repaired by screwing on new flanges, the port second intermediate cylinder on this ship had an extensive break. This was successfully repaired by arc-welding a cast-steel patch in place, also welding a jagged L-shaped break. The total length of weld was 127 inches. Steel studs were used to insure good connection between the cast iron and welded metal.

All cylinders requiring repairs were removed from the ship and welded in shop. This was deemed desirable both on account of easy welding and machining, and also so that complete inspection for flaws of any kind could be made on test. After welding the cylinders were bored and cover studs fitted in shop. Cylinder covers were then fitted, openings blanked off and test applied. It was found that when a cylinder under test was completely filled with water most of the leaks developed without further pressure. In every case, however, before the hydrostatic pressure was applied the water in the cylinder was heated by steam to at least 150 degrees F. The usual hydrostatic test pressure of about 50 per cent. excess of working pressure was then applied as follows:

High	300 pounds per square inch.
First intermediate	150 pounds per square inch.
Second intermediate	75 pounds per square inch.
Low	30 pounds per square inch.

After the hydrostatic test all cylinders were tested under steam, in the case of second intermediate and low-pressure cylinders to their working pressures, and, in the case of high-pressure and first intermediates, to 100 pounds.

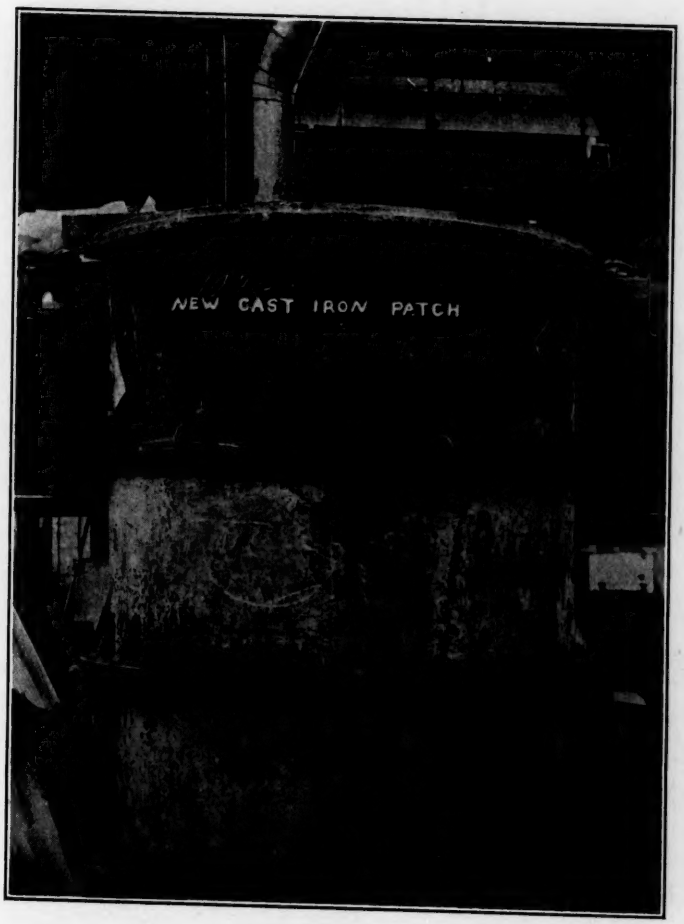


FIG. 1.

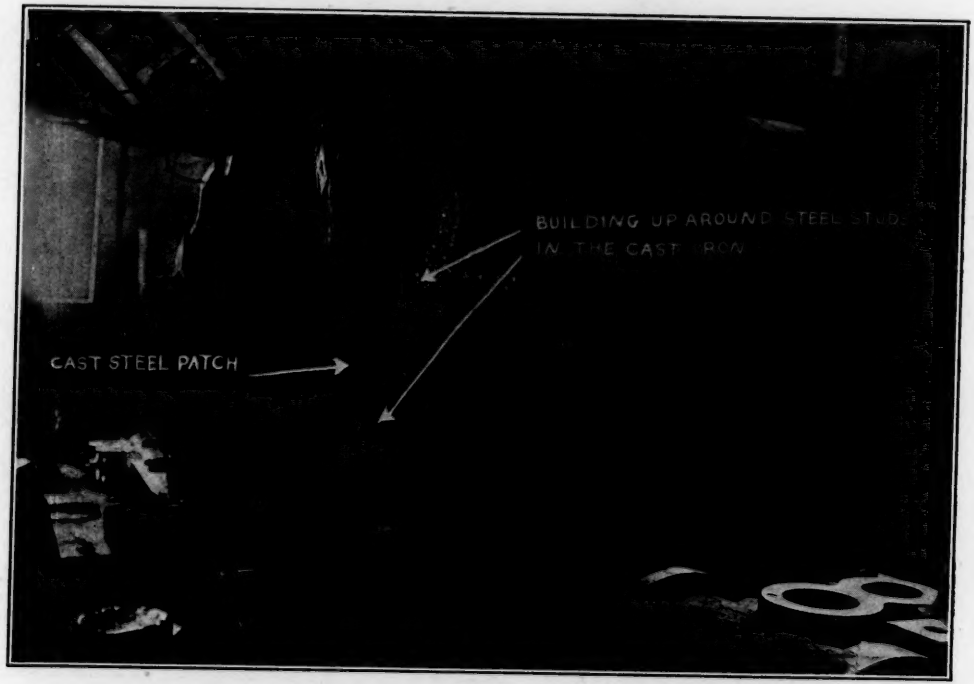

BUILDING UP AROUND STEEL STUDS IN THE CAST IRON
CAST STEEL PATCH

FIG. 2.

FIG. 3.

In none of the acetylene welds did any leaks, other than one or two pin holes, show. In the arc welds leaks of various sizes developed in practically all welds, and were closed by rewelding or caulking. The large weld in the *Bulgaria,* for example, was tested four times before passed as tight. In every case the leak was between the welded metal and the cast-iron cylinder wall. No leaks showed where the welded metal joined the cast-steel patch.

In preparing the cylinders for either arc or acetylene welding the jagged edge of the break was first smoothed off and then beveled to approximately 45 degrees; wooden patterns representing the patch were then fitted to the break in the cylinder and a cast-iron or steel patch made from this pattern. The cast-iron or cast-steel patch, after cleaning, was fitted to the cylinder and beveled to approximately 45 degrees on the welding edge. The patch was then clamped in place for welding, Fig. 1.

When arc welding was to be used all cast-iron edges adjacent to the weld were drilled and tapped for ½-inch studs spaced about 1½ inches center to center. Two rows of studs, staggered, were employed. See Fig. 2. The theory of using steel studs is that the weld will take hold of the studs without fail and thus assure a good hold on the cast iron. Studs were, of course, not fitted when acetylene welding was used.

With the patch secured in place the arc welding proceeded by first building up the edges of the V slot from one end to the other and then filling in between these steel-plated surfaces with the welding material. At no time during the weld was the adjacent metal allowed to become very hot. Overheating induces shrinkage or cooling strains in the weld. These are particularly noticeable where the weld joins the cast iron. The welding material is steel wire for all arc welds.

Acetylene welding is fundamentally different in application, and, while in arc welding every effort is made to keep the vicinity of the weld cool, in acetylene welding preheating is employed and the actual filling of the V slot is done in lengths

of about six inches. The sides of the slot are actually fused with the torch and small pieces of welding rod and flux added to the puddled metal. Having built up six inches flush, another section of the same length is then fused and filled. On heavy cast iron the weld proceeds at about eight inches per hour. By skillful manipulation of the torch and the use of flux most of the slag and other impurities are worked out of the weld. Cast-iron welding sticks are used for filling in.

In order to prevent after-cooling cracks in the metal near the weld it is necessary, in acetylene welding large bodies, to employ preheating. This is most satisfactorily done with a charcoal fire built under the break. In most of the cylinders this was started about four hours before the welding. The fire is kept going all the time during the welding and is allowed to die out in place when the weld is finished. In fact, when finished, the weld, fire and all, is covered up and slow cooling encouraged.

Another point required in acetylene welding is that due to the fluid condition of the metal when the torch is applied it is necessary to keep the surface of the metal horizontal. In a cylinder, therefore, when the break is circumferential, it is necessary to mount the cylinder on rockers or, in some other convenient way, make it possible to rotate it as the weld proceeds. Fig. 3 shows the charcoal grate and cylinder on rockers.

As the foregoing brief description is primarily to illustrate the actual welds and not to go into the relative merits of the two methods of welding, no further discussion on that point is offered. The most important conclusion is that complicated breaks in fifteen cylinders, besides many lesser parts, were successfully repaired, and that today three valuable ships are in service against a country that strove in vain to render these ships useless for months to come.

A SHORT-CUT METHOD OF CALCULATING THE DISTANCE RUN FROM THE ENGINE SPEED OF A SHIP.

By C. M. Reed, Instructor, U. S. N. A., Associate.

It was the writer's privilege to spend several weeks on one of the larger battleships this summer, in connection with the engineering work of the midshipmen's cruise. One detail that seemed noteworthy was the large amount of time and work connected with working up the distance run during each hour, to be entered in the log.

At the request of the assistant engineer officer, the method described below was devised, which the writer believes will enable the work to be taken over by the clerical force of the log office.

From the curve of r.p.m. *vs.* speed in knots, the knots per hour corresponding to every five revolutions per minute were tabulated, and a logarithmic scale of revolutions was plotted. A logarithmic scale of knots per hour was also plotted along the same line, so that the one line shows the relation between revolutions and speed. Some distance from, and parallel to the speed scale, a scale of minutes run, also logarthmic, was plotted with the same unit of length.

Midway between the two scales previously described a line was drawn parallel to them, on which was to be plotted the scale of distance traveled. It is obvious that if a ship runs for six minutes at the speed of 10 knots per hour the distance covered will be one knot. A line was drawn joining ten knots on the speed scale with six minutes on the time scale. The intersection of this line with the center line gave the location of the unit or starting point for the distance scale. From this mark as an origin, with a unit of length equal to one-half of that used for the speed and time scales, a scale of distance run was laid off in both directions.

In using the chart a line is drawn joining the r.p.m. with

the duration of the run in minutes. The intersection of this line with the center scale gives the distance run. The example plotted on the chart is for a 28-minute run at 180 r.p.m., giving a distance of 5.7 knots traveled.

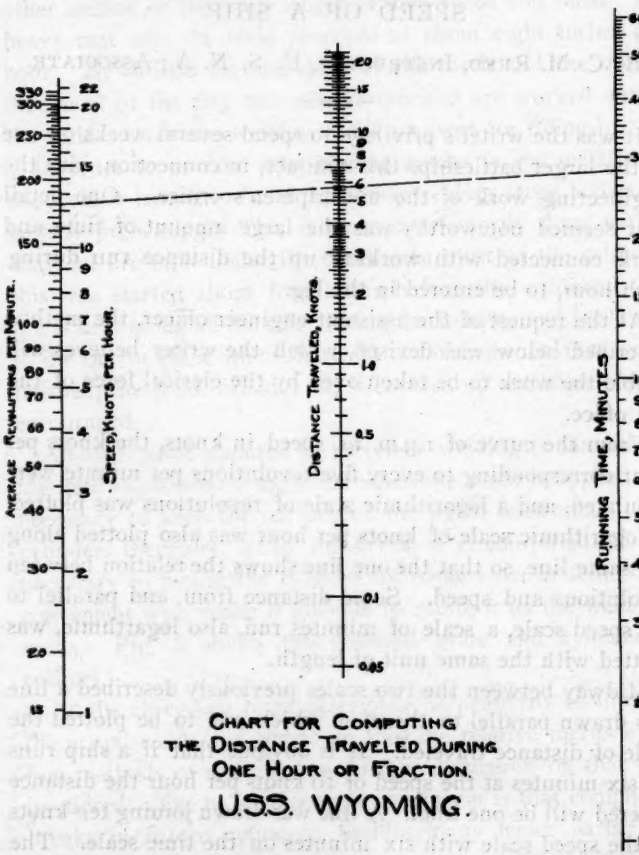


CHART FOR COMPUTING THE DISTANCE TRAVELED DURING ONE HOUR OR FRACTION.

U.S.S. WYOMING.

A similar chart could be plotted for any ship by locating the revolution scale properly with respect to the speed in knots scale, as determined by the speed curve for the ship in question. The logarithmic scales may be conveniently laid off with the aid of a slide rule, or a sheet of logarithmic cross-section paper.

AN IMPACT-ENDURANCE TESTING MACHINE.

By D. J. McAdam, Jr., Associate.

In the few repeated-impact machines now on the market there are certain defects, which have prevented their more general use in the testing of metals. As a result of attempts to correct some of the obvious defects in one of these machines at the U. S. Naval Engineering Experiment Station a new machine, free from such defects, has been developed. The new machine has been in use at the Station for some time, has been tested thoroughly, and found to give satisfactory results.

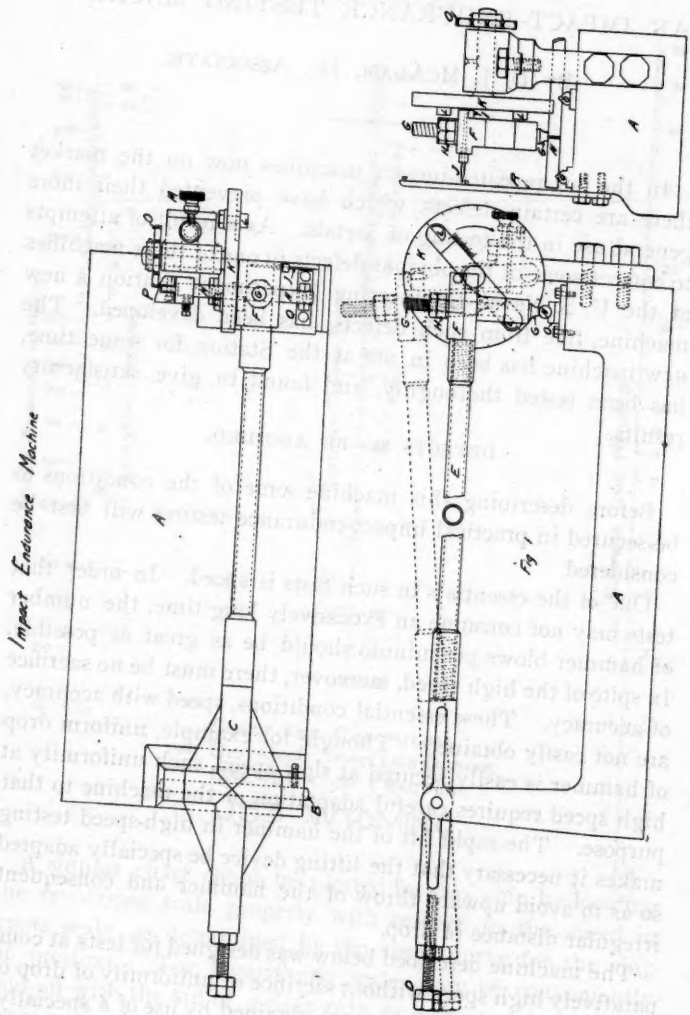
DEFECTS TO BE AVOIDED.

Before describing this machine some of the conditions to be secured in practical impact-endurance testing will first be considered.

One of the essentials in such tests is speed. In order that tests may not consume an excessively long time, the number of hammer blows per minute should be as great as possible. In spite of the high speed, moreover, there must be no sacrifice of accuracy. These essential conditions, speed with accuracy, are not easily obtained. Though, for example, uniform drop of hammer is easily secured at slow speed, such uniformity at high speed requires careful adaptation of the machine to that purpose. The rapid lift of the hammer in high-speed testing makes it necessary that the lifting device be specially adapted so as to avoid upward throw of the hammer and consequent irregular distance of drop.

The machine described below was designed for tests at comparatively high speed without sacrifice of uniformity of drop of hammer. These conditions are obtained by use of a specially-

designed hammer and lifting device, and by careful adjustment of the speed with the help of automatic diagrams.



DESCRIPTION OF MACHINE.

The machine is illustrated by the sketch, Fig. 1, the letters A to R indicating the various parts. The main divisions of the machine are: the hammer C to J, the lifting device K to M, and the specimen holder and connection P and Q.

The hammer is pivoted on the cylinder B, which is held in an upright arm of the base A. The hammer consists of a cast-iron piece, C, a steel tube, E, and the head F to J. The tube E provides for the necessary rigidity of the hammer without undue weight. The head of the hammer consists of the steel piece F fastened to the end of the tube E, the hardened tool-steel cylindrical piece G held to F by the lock nuts H, the hardened tool-steel piece I, and the pencil J. The weight of the hammer can be varied by addition of extra pieces, which are held between the part F and the lock nuts H. As the weight of the hammer is thus changed, adjustment of the bolt D keeps the center of percussion directly above the specimen N.

The hammer is lifted by the adjustable cams L, attached to the revolving disc K. The height of lift can be varied by setting the cams at various positions in a slot in the lifting disc. During a revolution of the disc as each cam comes in contact with the lower surface of I, the hammer is lifted; it is then released as the cam reaches the top of its path. The center of the axis of the disc K is so placed that it is below the lowest position of the contact surface of I by an amount equal to the radius of the cams L. By this arrangement, no matter what their position in the slot, the cams always start to lift the hammer when their upper surfaces are in a horizontal plane with the axis of K. The axis of K is adjustable in a horizontal plane by means of the screw M, so that the period of lift can be varied from slightly less to slightly greater than one-quarter revolution of the disc K. The disc K can be revolved at adjustable uniform speed, either by a chain drive and the sprocket wheel O, or by any other suitable gearing.

The specimen N is supported on two bearings Q, illustrated

in g eater detail in Fig. 2. A carefully machined cylindrical specimen with a nick of definite shape and depth is used. It may be rotated at uniform speed so that the hammer blows fall at regular intervals around the circumference, or else a device can be used for sudden turning of the specimen through 180 degrees after every one or two blows. The driving shaft is connected to N by any suitable flexible connection P, and the rotation is independent of that of the disc K. A revolution counter and automatic stop are attached to the machine, so that indication is given of the number of cycles necessary to break the specimen.

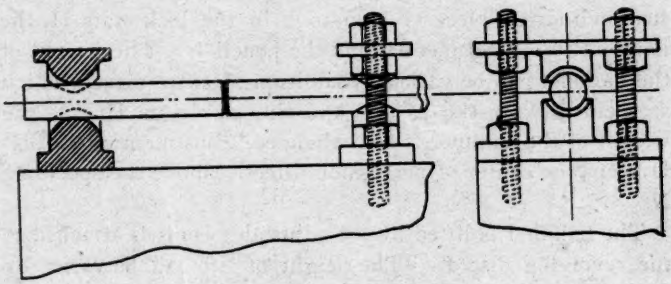


Fig. 2

PROCEDURE IN AN ENDURANCE TEST.

The cams having been set for a definite height of drop of hammer, the axis of lifting disc K must be adjusted in the horizontal plane so that the hammer will be released when each cam is near the top of its path. Theoretically it might seem that the cam should be exactly at the top of its path when the hammer is released. The hammer would then drop while the disc is turning through the next 90 degrees and the other cams would then be in position to lift the hammer again. Practically, however, it seems that better results are secured if the release occurs when each cam has passsed the highest point in its path by about 3 degrees to 5 degrees. The angle of revolution is thus 93 degrees to 95 degrees during the lift, and 85 degrees to 87 degrees during the drop of hammer.

The speed of the lifting disc K must now be properly adjusted. Theoretical considerations and experiments with this machine show that the speed should be such that the disc turns through about one-fourth revolution in the time required for the hammer to fall on the specimen from the highest point of its path. The time of drop, moreover, depends on the distance of drop under force of gravity. If h and t respectively represent distance and time of drop, and a represents the acceleration of the hammer, the relation between these three quantities may be expressed by the well known equation, $h = \frac{1}{2}at^2$. Since h is small compared with the distance between the centers of suspension and percussion of the hammer, a is nearly constant for all values of h, and its absolute value depends only on the moment of inertia and the position of center of gravity of the hammer. The time of drop, then, varies directly as the square root of the distance of drop of hammer. The proper speed of the disc consequently varies inversely as the square root of the distance of drop.

As the energy of hammer blow decreases, therefore, and the number of blows required to break the specimen increases, the succession of blows becomes more rapid. This is a great advantage in practical endurance testing.

For adjustment of conditions so as to secure proper speed and uniform drop of hammer the motion of the hammer may be analyzed by means of automatic diagrams. Such diagrams are made by the pencil J on a horizontally-moving strip of paper. The pressure of the pencil, regulated by a spring, is so slight that the motion of the hammer is not sensibly impeded. Nevertheless the diagram is not taken continuously, but is used only as a means of obtaining proper adjustment of the apparatus.

Figs. 4, 5 and 6 show typical diagrams for a definite distance of drop of hammer. In each diagram the portions a, b and c, respectively, represent the hammer's rebound from the specimen, its lift by the cam, and its drop again upon the specimen. Fig. 4 illustrates the motion of the hammer when the speed of the lifting disc is somewhat too slow. The

hammer has had time to rebound from the specimen and start downward again before the cam starts to lift it. Though this gives a fairly uniform drop of hammer, better conditions are obtained by slightly increasing the speed of the disc so that, as illustrated in Fig. 5, the hammer is at the top of its rebound when it is caught by the cam. With still further increase of speed, as illustrated by Fig. 6, the hammer is lifted by the cam immediately after its rebound from the specimen. Though

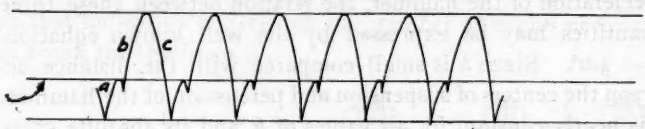


Fig. 4.

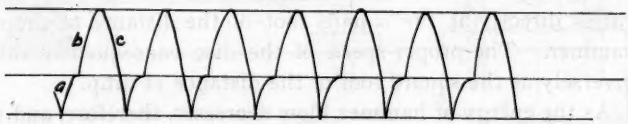


Fig. 5

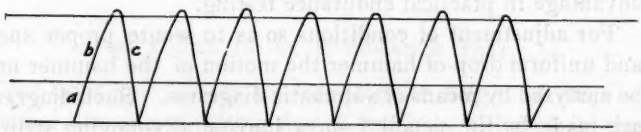


Fig. 6

it was expected that the latter would be the best adjustment, there is under these conditions always a slight upward throw of the hammer, resulting in irregular distance of drop. The best conditions, therefore, seem to be as illustrated by Fig. 5; with this arrangement a very uniform distance of drop is obtained throughout each test.

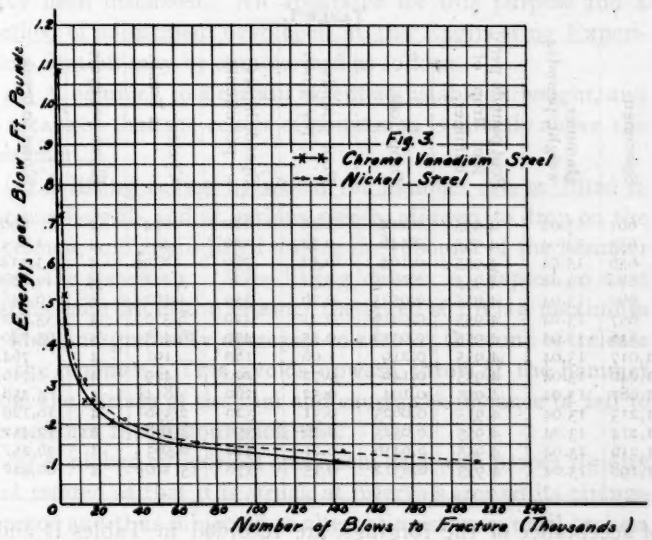
In the machine in use at the U. S. Naval Experiment Station the drop of the hammer can be varied from about 2.5 to

0.25 inches. Corresponding to this variation, the proper speed of the lifting disc varies from about 85 to 290 revolutions per minute. This means that the speed of hammer varies from about 170 to 580 blows per minute, or from about 10,000 to 35,000 blows per hour. The number of blows necessary to fracture the specimen will, of course, depend on the energy of a single blow and can be varied from only a few to many thousand.

TEST RESULTS OBTAINED.

With the above described machine tests have been made on various kinds of steel in order to determine their relative endurance qualities. It is proposed also to extend this method of testing to non-ferrous metals.

The results of a series of tests on two kinds of steel are given in Table I. The word cycle as used in this table means one complete revolution of the test specimen. Since, in the machine now in use, the specimen is rotated suddenly through 180 degrees after every two blows of the hammer, one cycle



corresponds to four blows of the hammer. The values in the third column of Table I are obtained by multiplying the values in the second column by the ratio of the distances from the center of suspension of the hammer to its center of gravity and center of percussion. In Fig. 3 the values for energy of impact are plotted against the number of blows required to fracture the specimen.

These results are here given merely for the purpose of illustrating the usefulness and range of the machine. They show the endurance qualities of two samples of metal that were available. The results, however, are not intended to show the relative merits of nickel and chrome-vanadium steel.

The test specimens were taken from the test ends of two forgings now in use as battleship shafting. Specimens numbered 620 to 653 are from a forging of annealed nickel steel; specimens 1017 to 1216 are from a forging of annealed chrome-vanadium steel. The chemical composition and the average of the tension-test results given by the inspector at the time

TABLE I.

Specimen.	Hammer weight, pounds.	Foot pounds per foot drop of hammer.	Drop, foot.	Foot pounds per blow.	Speed, blows per minute.	Cycles to break.	Blows per cycle.	Blows to break.
661	13.04	4.955	0.219	1.08	180	174	4	696
653	13.04	4.955	0.146	0.72	220	401	4	1,604
656	13.04	4.955	0.104	0.52	260	833½	4	3,334
620	13.04	4.955	0.0625	0.31	320	3,222	4	12,888
660	13.04	4.955	0.0525	0.26	350	4,162	4	16,648
657	13.04	4.955	0.0416	0.21	420	8,377	4	33,508
636	13.04	4.955	0.0312	0.15	470	16,435	4	65,740
1,017	13.04	4.955	0.219	1.08	180	191	4	764
1,040	13.04	4.955	0.146	0.72	220	429	4	1,716
1,087	13.04	4.955	0.104	0.52	260	814½	4	3,258
1,213	13.04	4.955	0.0625	0.31	320	2,530	4	10,120
1,214	13.04	4.955	0.0525	0.26	350	3,108	4	12,432
1,216	13.04	4.955	0.0416	0.21	420	6,563	4	26,252
1,198	13.04	4.955	0.0312	0.15	470	15,110½	4	60,442

of acceptance of the forgings, are recorded in Tables II and III below.

TABLE II.

Material.	C	Mn	S	P	Ni	Cr	V
Nickel steel.....	0.42	0.54	0.021	0.011	3.27	...	...
Chrome-vanadium steel.....	0.41	0.78	0.038	0.008	...	0.85	0.14

TABLE III.

Material.	Tensile strength, pounds per square in.	Yield point, pounds per square inch.	Elongation, per cent. in 2 inches.	Reduction of area, per cent.
Nickel steel.....	89,100	52,500	26 2	48.1
Chrome-vanadium steel..	82,200	52,700	34.2	63.4

SUMMARY.

Conditions necessary for practical impact-endurance testing have been discussed. An apparatus for this purpose and a method of adjustment devoloped at the Engineering Experiment Station may be summarized as follows:

(*a*) A hammer, to a certain extent adjustable in weight, and so arranged that the center of percussion is directly above the specimen.

(*b*) A lifting device by which the hammer can be lifted to various heights and at various speeds, allowed to drop on the specimen, and again lifted during the rebound of the hammer from the specimen. The lifting device is adapted so that during each lift of the hammer the speed of lift is a maximum at first and gradually decreases to zero at the time of release of the hammer. This avoids upward throw of the hammer from the surface of the cam and causes the distance of fall to be uniform throughout each test.

(*c*) A nicked cylindrical specimen, held with proper rigidity and rotated so that it is struck at intervals around its circumference and thus subjected to alternating stresses until broken.

(*d*) A method of determining and regulating the proper

speed of revolution of the lifting disc K so that each cam begins to lift the hammer at the proper instant during the rebound of the hammer from the specimen. This secures uniform distance of the drop of hammer and, consequently, uniform energy of blow.

NOTE.—The writer wishes to acknowledge the valuable assistance of Mr. L. E. Foster, who is employed in the metals laboratory of the Station, in working out details of this machine.

CHARACTERISTICS AND SPECIFIC SPEEDS OF CENTRIFUGAL PUMPS.

F. G. HECHLER, M. E.

The present increasingly extensive use of centrifugal pumps would seem to warrant a consideration of their operating characteristics and the development of the "specific speed" method of classifying them and predicting their probable performance under various conditions.

For handling large quantities of water under low heads, as in drainage work and for unwatering cofferdams, the use of centrifugal pumps has become almost universal, their advantages under these conditions being so great as to practically preclude even a consideration of any other type. Under other conditions, however, where relatively high heads and small quantities are required, their adoption has not been so general. But during the last eight or ten years nearly all of the large central power stations have adopted the centrifugal boiler-feed pump as their standard equipment. Marine practice has been more conservative in this regard, and it is only recently that there has been any decided tendency in this direction.

The centrifugal pump possesses inherent advantages that make it the ideal unit for boiler feeding, or any other service, provided it is properly chosen for the work it is to do. The freedom from shock and water hammer, due to the steady discharge, the absence of all danger of building up an excessive pressure that might cause breakage in the pipe line if all valves on the discharge line should be closed, the elimination of the necessity of air chambers and relief valves, and the simple lubrication requirements are all decided advantages.

With low and medium heads the simple and inexpensive single-stage volute pumps give excellent satisfaction with reasonably good efficiencies. But for high pressures, necessary with boiler-feed pumps, it is impracticable to use single-stage pumps unless very high speeds are used. With high heads and small quantities, moreover, leakage through the clearance spaces at the periphery of the impeller becomes proportionately large. For these reasons the multistage pump has been developed; the number of stages used depending on the type of pump and on the head pumped against. Usually the limiting head per stage is about 100 to 150 feet, though tests in which the head per stage was 375 feet have been reported. Such pumps, operating at high heads and high speeds with small quantities, necessarily have small dimensions, and it is imperative that the calculated dimensions be strictly adhered to in the construction, or the finished pump will not give the anticipated results because of excessive leakage.

Water enters centrifugal pumps at the center or "eye" of the impeller. Passing radially outward along the blades it is discharged into the pump casing. The impeller increases both the pressure and the velocity of the water, the velocity term being the greater of the two. It is necessary to change this velocity into pressure in the pump casing, and the efficiency of the pump depends largely on the success with which this is accomplished.

As shown in Fig. 1, the water, as it leaves the impeller and enters the pump case, is moving with a velocity w_2 relatively to the pump case. In order to decrease this velocity, and thus increase the pressure head, it is the usual practice with high-head pumps to make use of diffusing vanes as shown in Fig. 1. Obviously these vanes must be so shaped that the water as it leaves the impellers will enter without shock or eddy, that is, the entrance edge of the vane must be tangent to the direction of the water at exit from the impeller. The diffusing vane should be smooth, the entrance edge should be

sharp and there should be no sudden changes in direction or in cross-sectional area.

The first condition, that the water should enter the diffuser without shock, can be fulfilled only for one rate of discharge if the pump is running at constant speed. Referring to the exit velocity diagram in Fig. 1, at constant speed u_2, the linear velocity of the impeller in feet per second remains constant; v_2 the velocity of the water relatively to the im-

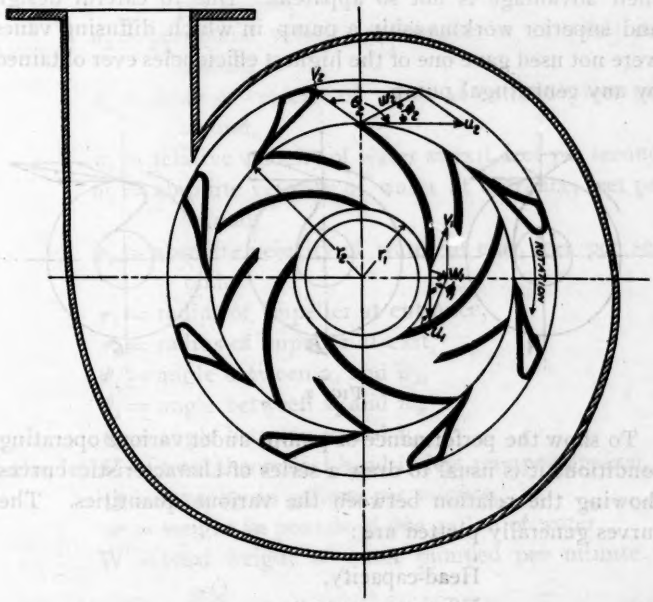


FIG. 1.

peller in feet per second, depends on the quantity, and, consequently, w_2, which is the absolute velocity of the water leaving the impeller, and the resultant of v_2 and u_2, will change in direction and magnitude as the quantity of water changes, resulting in consequent losses due to shock and eddies. To meet this difficulty adjustable diffusers have been proposed, but have met with little success because of the

added complication. Another method of maintaining the proper angular relations would be to change the speed of the unit so that u_2 changes proportionally to v_2, but this may cause a change in the head produced by the pump, which may not be desirable or even permissible. In general, for high heads, the use of diffusing vanes is to be recommended, as, with them the loss due to shocks and eddies is probably less than it would be if they were omitted. In low and medium heads their advantage is not so apparent. Due to careful design and superior workmanship a pump in which diffusing vanes were not used gave one of the highest efficiencies ever obtained by any centrifugal pump.

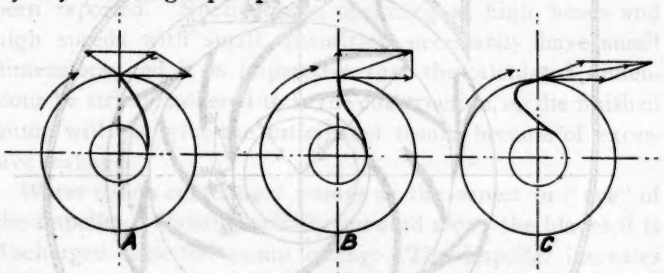


FIG. 2.

To show the performance of pumps under various operating conditions it is usual to draw a series of characteristic curves showing the relation between the various quantities. The curves generally plotted are:

Head-capacity,
Efficiency-capacity,
Brake horsepower-capacity,
All at constant speed.

For economical operation the efficiency-capacity curve should not only reach a high value but should be as flat as possible, so that the economy will be good over as wide a range of capacities as possible.

The head-capacity curve may, by proper design of impeller, be made to assume almost any desired shape. Fig. 2 shows

three designs, A, B and C, for the shape of impeller blades. A shows the blades at exit bent backward, B shows radial blades, and C shows them bent forward. The general shape of the head-capacity characteristic that each of these designs will give can be readily found. For simplicity consider the ideal case, neglecting friction and all losses due to shock and eddies. Referring to Fig. 1,

Let u_1 = peripheral velocity of impeller at entrance, feet per second,
u_2 = peripheral velocity of impeller at exit, feet per second,
v_1 = relative velocity of water at entrance, feet per second,
v_2 = relative velocity of water at exit, feet per second,
w_1 = absolute velocity of water at entrance, feet per second,
w_2 = absolute velocity of water at exit, feet per second,
r_1 = radius of impeller at entrance,
r_2 = radius of impeller at exit,
Φ_1 = angle between u_1 and w_1,
Φ_2 = angle between u_2 and w_2,
θ_2 = angle between u_2 and v_2,
also let H = total theoretical head in feet pumped against,
Q = capacity in gallons per minute,
w = weight in pounds of one gallon of water,
W = total weight of water pumped per minute = wQ.

The energy supplied to the pump (since the efficiency is assumed to be 100 per cent.) is all used in producing the theoretical head H. The work done is evidently wQH foot-pounds.

At entrance to the impeller the impulse of the entering water is $\frac{WV}{g} = \frac{wQw_1}{g60}$, and its component in the direction of motion of the impeller at entrance is $\frac{wQ}{60g} \times w_1 \cos \Phi_1$. Sim-

ilarly, at exit there is a force, tending to retard the motion of the impeller, of $\frac{wQ}{60g} \times w_2 \cos \Phi_2$.

Hence the work, which is force times distance, is given by the equation

$$\frac{wQ}{60g}\left(w_2 \cos \Phi_2 u_2 - w_1 \cos \Phi_1 u_2 \times \frac{r_1}{r_2}\right) = \frac{wQH}{60},$$

or $$\frac{1}{g}(w_2 \cos \Phi_2 u_2 - w_1 \cos \Phi_1 u_1) = H.$$

The water usually enters the impeller radially, so that $\Phi_1 =$ 90 degrees and $u_1 w_1 \cos \Phi_1 = 0$;

hence $$H = \frac{1}{g} u_2 w_2 \cos \Phi_2.$$

From the exit velocity triangle we have

$$v_2^2 = u_2^2 + w_2^2 - 2u_2 w_2 \cos \Phi_2.$$

Eliminating cos Φ_2 from the expression for H,

$$H = \frac{1}{g} u_2 w_2 \left(\frac{w_2^2 + u_2^2 - v_2^2}{2u_2 w_2}\right) = \frac{w_2^2 + u_2^2 - v_2^2}{2g}; \quad (A)$$

also $$w_2^2 = v_2^2 + u_2^2 + 2v_2 u_2 \cos \theta;$$

whence $$H = \frac{v_2^2 + w_2^2 + 2v_2 u_2 \cos \theta + u_2^2 - v_2^2}{2g} = \frac{u_2^2 + v_2 u_2 \cos \theta}{g}. \quad (B)$$

For a given speed and quantity, u_2 and v_2 are fixed, and the effect of a change in the angle θ on the head H can be determined. With forward-tipped blades cos θ is positive, so that from equation (B) $H > \frac{u_2^2}{g}$; for radial blades cos θ is zero, so that $H = \frac{u_2^2}{g}$; and for backward-tipped blades cos θ is negative, so that $H < \frac{u_2^2}{g}$. If the speed is kept constant and the

quantity is increased v_2 increases, so that with forward-tipped blades H increases with the quantity; for radial blades H remains constant; and for backward-tipped blades H decreases as the quantity increases. For the ideal conditions we have assumed, the curves have the shape shown by solid lines in Fig. 3. The shut-off, or no delivery, pressure would be the same in each case, for with no water flowing through the impeller $v_2 = 0$, and $w_2 = 0$, so that $H = \frac{u_2^2}{2g}$. In the actual pump, friction and other losses modify these curves to varying degrees, so that the shapes shown dotted in the same figure are approximated. Each of these types meets certain requirements most effectively.

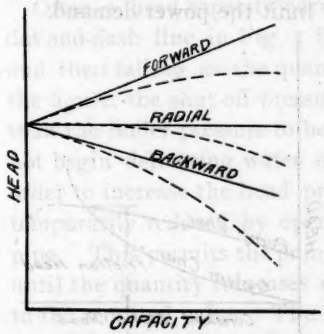


FIG. 3.

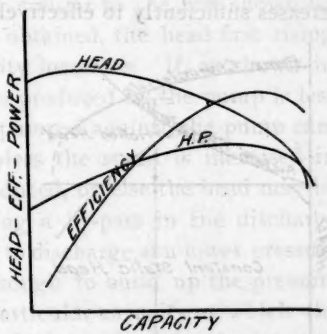


FIG. 4.

In equation A, the term $\frac{w_2^2}{2g}$ represents the velocity head of the water as it leaves the impeller. For efficient operation this must be largely converted into a pressure head by using either diffuser vanes or a volute casing. This term has its maximum value with forward-tipped blades, and its minimum value for backward-tipped blades. Hence, other things being equal, the backward-tipped blade impeller should give the greater efficiency. Most commercial pumps have vanes of the backward-bent type. This type possesses the added advantage, due to the dropping head-capacity characteristic, of

automatically limiting the power required as the capacity increases, so that there is no danger of overloading the motor; in fact, it is quite common to find that as the capacity increases somewhat beyond the rated value the power required actually decreases. This makes it possible to choose a motor of the proper size to operate normally at its rated load and maximum efficiency over a wide range of capacities. Such characteristics are shown in Fig. 4. Note that the brake horsepower does not vary greatly from the full load value, so that the motor driving the pump, whether it is an electric motor or a steam turbine, may operate at almost constant efficiency over the whole range. Also the design is such that after the rated capacity is exceeded the head developed decreases sufficiently to effectively limit the power demand.

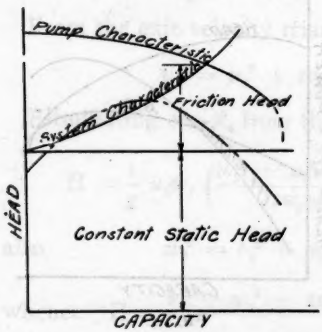


FIG. 5.

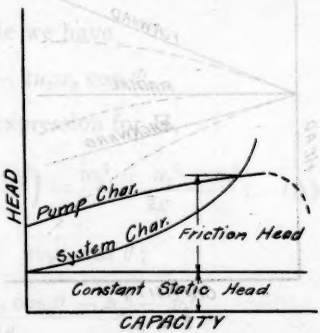


FIG. 6.

For some uses though, it is not desirable or even possible to permit the head to decrease to any great extent as the capacity is increased. Such a condition is met in boiler feeding. A certain minimum pressure must be maintained in order to force the water into the boiler, and also the pump motor should be able to run at a constant speed under the control of a governor without building up an excessive pressure even if all of the valves on the discharge line should be closed. In boiler feeding the resistance to be overcome is made up of two parts, as shown in Fig. 5. In this case the

constant static head equal to the boiler pressure forms the greater part of the resistance, the remainder is the friction head, which is proportional to the quantity, and hence to the square of the velocity, and depends on the length of the discharge pipe, its condition, the number of valves, elbows, etc. It is evident that in this case the power required increases with the capacity, and to prevent an excessive overload on the motor it is necessary to so design the impeller that the head-capacity curve will drop suddenly, as shown by the dotted extension of the curve, soon after the capacity exceeds the maximum desired. This permits the use of a motor of the proper size to run at its best efficiency for normal operation.

Often a head-capacity curve similar to the one shown by dot-and-dash line in Fig. 5 is obtained, the head first rising and then falling as the quantity increases. If, as shown in the figure, the shut-off pressure produced by the pump is less than the boiler pressure to be pumped against, the pump cannot begin delivering water unless the speed is increased in order to increase the head produced, or else the head may be temporarily reduced by opening a by-pass in the discharge pipe. This permits the pump to discharge at a lower pressure until the quantity increases enough to build up the pressure to the required value. The particular capacity at which the pump is operating at any time is determined by the intersection of the two head-capacity characteristics—one for the pump, the other for the system into which the pump is discharging. By opening or closing valves in the feed line the system characteristic is changed, giving a different point of intersection and hence a new operating point for the pump.

If the load on the pump is largely a frictional one, as in circulating pumps for condenser service, the head produced by the pump should increase with the capacity in order to offset the increase in friction with the increase in capacity. This is best done by using a pump with the impeller blades bent forward, giving the rising head characteristics shown in Fig. 6. In this case the load on the driver increases rapidly

as the capacity increases, so that it is necessary to provide some means of limiting the maximum demand. This can be accomplished as before by making the head characteristic drop off abruptly, as shown by the dotted extension.

TEST CHARACTERISTICS.

If the complete performance of a centrifugal pump is desired it must be tested at various speeds, and at each speed the head, capacity, power imput and the efficiency must be determined for all capacities from shut-off to the maximum discharge. These results are best shown in the form of curves, as in Fig. 7, which shows the head, the horsepower input, and the efficiency as functions of the capacity for an actual pump at various speeds from 700 to 2,000 revolutions per minute. The lowest speed at which good efficiencies are attained is 1,400 revolutions; at this speed the pump has an efficiency of 55 per cent. when pumping 160 gallons per minute at a head of 170 feet. It will be observed that the pump has practically the same maximum efficiencies at speeds up to 1,800 revolutions, but that at each speed it will be operating at a different capacity and head.

In Fig. 8 the head-capacity curves have been redrawn, and on each curve the small circles represent points of constant efficiency obtained by cross plotting from Fig. 7. Curves drawn though these points are lines of constant efficiency and are shown solid in the figure. In addition a series of parabolas has been drawn. The parabolas are shown by the light dash lines, and it is to be noted how closely they agree with the heavy iso-efficiency lines over the usual operating range of the pump, that is, to a point just beyond the point of maximum efficiency. The line drawn through the maximum efficiency points at each speed is almost identical in form with the parabola near it. At times the iso-efficiency curves are drawn as ovals, as shown by joining the ends of the two curves for 50 per cent. efficiency.

Such a method of connecting the points of constant effi-

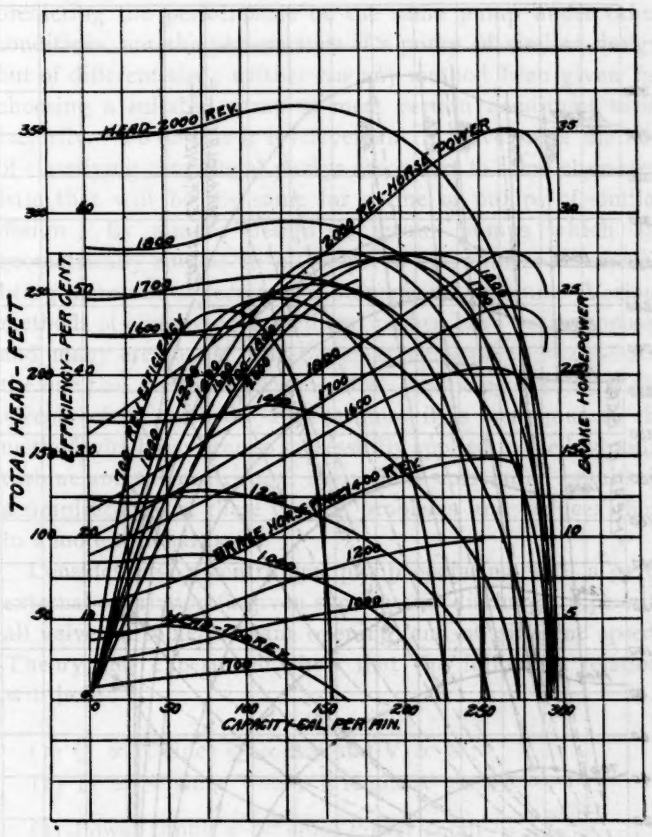


FIG. 7.

ciency directs attention forcibly to the most efficient range of operation for which the pump is adapted and particularly to the fact that it is not economical to use the pump at a speed either too low or too high. However, Fig. 8 seems to indicate that these constant-efficiency lines to the left of the line of maximum efficiency are not ovals, as their curvature is in the wrong direction. It is clear that a centrifugal pump may have the same efficiency when operating at different speeds,

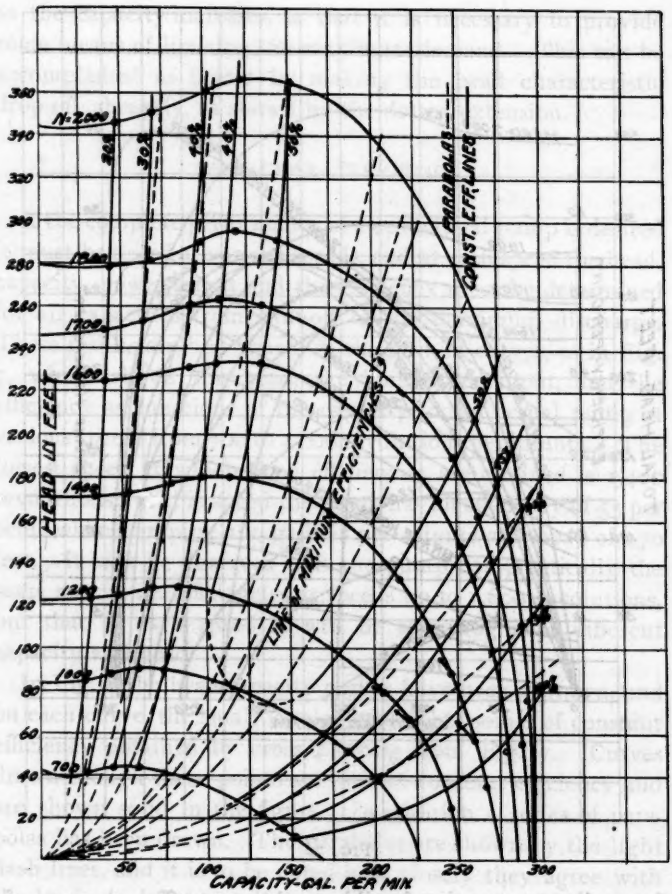


FIG. 8.

providing the head and capacity are the proper ones corresponding to the speed. The question of the relations of these factors will now be taken up under the general head of Specific Speeds.

SPECIFIC SPEEDS.

The characteristic curves so far considered show completely the performance of the pump tested, but give no method of

predicting the performance of the same pump under other conditions, nor the performance of a pump of similar design but of different size; neither has any method been given for choosing a suitable pump to meet certain conditions satisfactorily. To do this it is convenient to have some method of classifying centrifugal pumps according to some characteristic that will be the same for a line of pumps of similar design. By similar design is meant pumps which are geometrically similar in all respects, every linear dimension being increased or decreased a proportional amount. Various methods of classifying centrifugal pumps have been proposed and many are in use, but the most convenient method and the one that seems to give the most promising results is the *specific speed* method. Furthermore it is analogous to the method which has been so successfully applied to the hydraulic turbine and the centrifugal fan. This correlation results in a simplification of these various problems and reduces them to a more rational basis.

Consider first a centrifugal pump operating with a given external restriction (a given suction and discharge pipe with all valves at some constant opening) and at different speeds. Theory and experiment show that the following relations will hold:

(1) $Q \propto N$ since $Q \propto AV$ and $V \propto N$;

(2) $H \propto N^2$ since $V = \sqrt{KH}$ and $V \propto N$;

(3) Power Input $\propto N^3$ since Power Input $= \dfrac{wQH}{\text{Efficiency}}$;

(4) Efficiency constant.

Though the efficiency is assumed to be constant this is true only to a limited extent. An examination of a complete set of test results, as in Fig. 8, will show that the efficiency remains constant only over a somewhat limited range.

Now consider a series of geometrically similar pumps with geometrically similar connected systems, operating at a constant number of revolutions per minute and at constant

efficiency, but having different impeller diameters. All parts of the different pumps and their connected systems must vary in the same ratio as the impeller diameters. In this case we can write:

(5) $Q \propto D^3$ since $Q \propto AV$; $A \propto D^2$ and $V \propto D$;
(6) $H \propto D^2$ since $D \propto V \propto \sqrt{KH}$;

(7) Power Input $\propto D^5$ since Power Input $= \frac{wQH}{\text{Efficiency}}$;

(8) Efficiency constant, revolutions constant.

If now the speed and diameter are both varied, the *efficiency remaining constant*, we obtain, by combining equations 1 and 5; 2 and 6; 3 and 7, the following relations:

(9) $Q \propto ND^3$ or $Q = K_1ND^3$;
(10) $H \propto N^2D^2$ or $H = K_2N^2D^2$;
(11) Power Input $\propto N^3D^5$ or Power Input $= K_3N^3D^5$;
(12) Efficiency constant.

The various terms in the above equations are defined as follows:

H = Total head in feet pumped against;
Q = Capacity in gallons per minute;
N = Revolutions per minute;
D = Diameter of impeller in inches;
A = Area in square feet of water passages through the impeller and normal to the direction of flow at point where the velocity of the water is V feet per second;
w = Weight of one gallon of water in pounds;
K_1, K_2 and K_3 are constants.

Equations 9, 10 and 11 lead at once to some interesting general conclusions regarding centrifugal pumps. Suppose, for instance, it is desired to double the quantity of water

pumped keeping the head constant; constant head requires constant peripheral velocity for the impeller, that is

$$\frac{\pi DN}{60} = \text{constant or } D \propto \frac{1}{N}.$$

From equation (9)

$$Q = K_1 N D^3 = K_1 N_1 \frac{K'}{N_1^3} = \frac{K}{N_1^2} \text{ or } N_1 = \sqrt{\frac{K}{Q}};$$

similarly

$$2Q = \frac{K}{N_2^2} \text{ or } N_2 = \frac{1}{1.41}\sqrt{\frac{K}{Q}}.$$

Hence the new pump must run at a speed $\frac{1}{1.41}$ times that of the original unit and the diameter of the impeller must be 1.41 times that of the first. It is essential to remember that all the dimensions of the pump must be changed in the same ratio as the diameter for this relation to hold.

If it is desired to increase the head H it may apparently be accomplished with equal facility by increasing either N or D, as equation 10 shows that $H = K_2 N^2 D^2$. But the effect on the capacity and the power required would be quite different in the two methods. For example, doubling the speed would double the capacity and require 8 times the power, while doubling the diameter would give 8 times the capacity and would require 32 times the power. It is well to note that these relations furnish a convenient method of correcting test results to a standard basis, as will be shown later.

Equations 9 and 10 may be combined by eliminating D, thus:

$$\frac{Q}{K_1 N} = D^3 = \frac{H^{\frac{3}{2}}}{K_2^{\frac{3}{2}} N^3},$$

$$N^2 = \frac{K}{K_2^{\frac{3}{2}}} \frac{H^{\frac{3}{2}}}{Q},$$

or

$$N = K_4 \frac{H^{\frac{3}{4}}}{Q^{\frac{1}{2}}}.$$

This equation shows the effect on the speed of a variation in the head and capacity when the efficiency remains constant. Considering the special case where H = one foot and Q = one gallon per minute, we have N = K_4, or K_4 is the speed in revolutions per minute of an impeller that will pump one gallon per minute against a head of one foot. This speed will be designated the specific speed and will be represented by N_s, so that

$$N_s = N \frac{Q^{\frac{1}{2}}}{H^{\frac{3}{4}}}, \tag{12}$$

To determine the diameter of the impeller to be used eliminate N from equations (9) and (10).

$$\frac{Q}{K_1 D^3} = N = \frac{H^{\frac{1}{2}}}{K_2 D}; \quad D^2 = \frac{K_2 Q}{K_1 H^{\frac{1}{2}}},$$

or

$$D = K_5 \frac{Q^{\frac{1}{2}}}{H^{\frac{1}{4}}} = D_s \frac{Q^{\frac{1}{2}}}{H^{\frac{1}{4}}}. \tag{13}$$

As before, if we make Q = one gallon per minute and H = one foot head we have D = K_5, or K_5 is the diameter of the impeller that runs at N_s revolutions per minute and pumps one gallon per minute against a head of one foot. This constant will be designated as D_s and may be termed the "specific diameter" of the impeller.

The power required by the "unit" impeller, which has a diameter D_s inches and runs at N_s revolutions per minute, may be obtained by combining either equations (9) and (11), or (10) and (11). Using (10) and (11) and eliminating N we obtain:

$$\frac{Q^3}{K_1^3 D^9} = N^3 = \frac{\text{Power}}{K_3 D^5},$$

$$\text{Power} = \frac{K_3}{K_1^3} \frac{Q^3}{D^4} = \frac{K_3}{K_1^3 D_s^4} \frac{Q^3 H}{Q^2} = K_6 Q H,$$

or making an allowance for efficiency to obtain the power input

$$\text{Power input} = \frac{K_6 Q H}{\text{Efficiency}} = P_s Q H, \tag{14}$$

where P_s may be called the "specific power" when Q = one gallon per minute and H = one foot head.

For certain problems it will be useful to determine another factor, which may be called the "Specific Tip Speed." The speed of the tips of the impeller blades in feet per minute is given by

$$V = \frac{\pi DN}{12}.$$

If for D and N we substitute D_s and N_s we must also replace V by V_s.

Then $$V_s = \frac{\pi D_s N_s}{12} = \frac{\pi}{12} \frac{DH^{\frac{1}{2}}}{Q^{\frac{1}{2}}} \times \frac{NQ^{\frac{1}{2}}}{H^{\frac{3}{4}}},$$

$$\text{or } V_s = \frac{\pi}{12} \frac{ND}{H^{\frac{1}{4}}}. \tag{15}$$

Here V_s is the speed the tips of the impeller must have to deliver one gallon per minute under one foot head.

From the foregoing it is evident that test results for a pump of a given type are necessary in order to make the above calculations. In order to make a complete analysis the data should include the speed, capacity, head, efficiency, diameter of the impeller and the type of pump. As has been previously noted, there are two general types of centrifugal pumps; namely, the volute type in which no diffusing vanes are employed and the turbine pump in which diffusing vanes are used. Each of these broad types might be subdivided into pumps having single or double suction, whether single or multistage, or whether it is a single, bi- or tri-rotor pump. But essentially these are merely modifications of the two fundamental types and bring in no new variables. Thus the purpose of multistaging is to increase the total head produced, and as the formulae here developed refer only to the head produced in one stage it is necessary in a multistage pump to divide the total head by the number of stages before substituting in the formulae. Similarly when the quantity of water

to be pumped is very large two, and sometimes three, impellers may be mounted on one shaft and operate in parallel. In this case the total capacity must be divided by the number of impellers to get the proper value for use in the formulae. It should be kept clearly in mind that the formulae as here given refer always to the head and quantity for one stage of a multistage pump, or for one impeller of a pump having several impellers operating in parallel.

To illustrate the application of the formulae to an actual problem the test data for a single-stage volute pump with an impeller 15½ inches in diameter are given in the first four columns of Table I. The remaining columns show the arrangement of the work for convenient tabulation of the desired results, which have been computed from the formulae already given.

TABLE I.

Speed revolutions per minute.	Q gals. per min.	Head per stage. Feet.	Efficiency.	$Q^{\frac{1}{2}}$	$H^{\frac{1}{2}}$	$H^{\frac{1}{4}}$	N_s	D_s	V_s
1,000	220	64	40	14.8	2.8	21.6	689	2.95	507
1,000	315	65	50	17.8	2.9	23.4	771	2.49	503
1,000	430	67	60	20.7	2.9	23.4	885	2.14	495
1,000	610	67	70	24.7	2.9	23.4	1,052	1.80	495
1,000	760	64	75	27.6	2.8	21.6	1,279	1.59	507
1,000	1,100	52	75	33.2	2.7	19.5	1,710	1.26	510
1,000	1,280	46	70	35.8	2.6	17.7	2,025	1.13	598
1,000	1,560	33	60	39.5	2.4	13.8	2,870	0.94	706
1,000	1,690	26	50	41.1	2.3	11.5	3,570	0.85	794

This unit was also tested at speeds of 1,200 and 1,400 revolutions per minute and the values of N_s computed for comparison with the above The results are shown in Fig. 9, curve A, for volute pumps. The efficiency-specific speed curve for 1,200 revolutions is shown solid, while those for 1,000 and 1,400 revolutions are shown as light dash lines. Up to the point of maximum efficiency the curves agree quite well, and show that within the probable limits of accuracy of the test data the specific speed is a constant at constant efficiency, as has already been stated.

All the available test data for centrifugal pumps have been tabulated for both the volute and the turbine types. To show the relation between the specific speed and efficiency, the specific speed has been calculated from these data for the maximum efficiency in each case and the values plotted as shown in Fig. 9, where the circles represent these individual N_s efficiency values. In both figures a smooth curve has been drawn to show the maximum probable efficiency that may be expected from each type of pump when operating under favorable conditions. In addition, the curves A, B, C, D, E, F, G, H and I show the variation of the specific speed and efficiency for certain individual pumps of each type calculated from the test data. Each of these curves shows the characteristics of all other pumps of homologous design. With both volute and turbine pumps it is evident that low specific speeds cannot be used if high efficiencies are desired. Low specific speeds mean either a low actual speed or a small capacity for the head produced. It is also possible to use lower specific speeds with turbine pumps than with volute pumps and still operate efficiently. At high specific speeds the volute pumps seem to have the better efficiency, as shown by the curves. High specific speeds are obtained either by using high rotative speeds or by making Q large compared with H. The latter method is the particular field in which volute pumps excel, namely, very large capacities under low heads. In this case if the capacity is very large it may be necessary to use a low rotative speed in order to keep the specific speed down to a reasonable value. In many cases, however, high specific speed designs have been developed in order to increase the rotative speed of the unit so that more efficient operation can be obtained when a motor or a steam turbine is used with a direct-connected drive. Fig. 10, shows the relation between specific diameter, D_s, and specific speed, N_s, for the same pumps. It is interesting to note how closely they fall to each other. In the same figure the relation between the specific tip speed V_s and the specific speed N_s is also shown. Fig. 10, might be still further amplified by crossplotting the efficiencies from

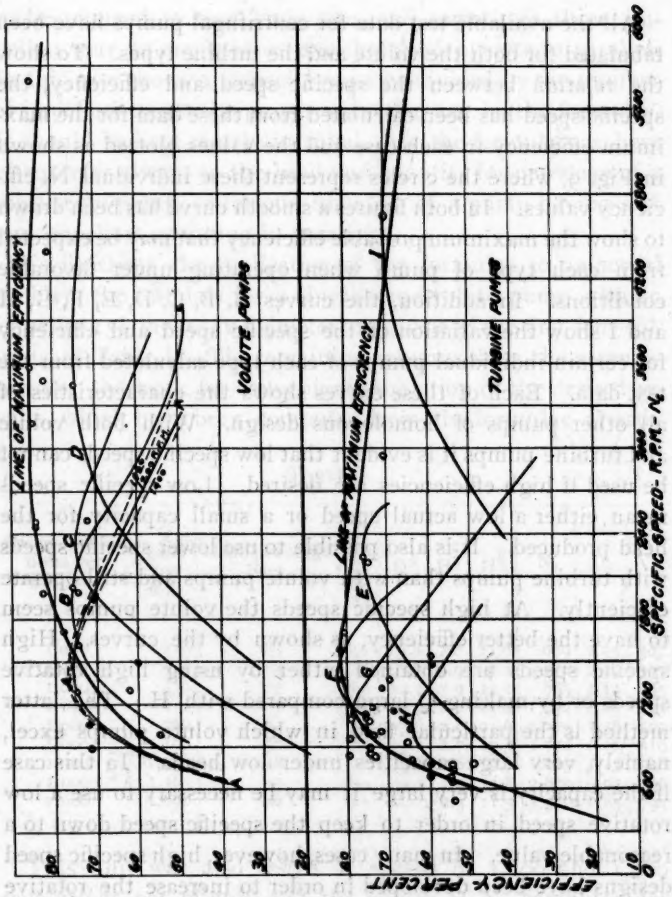


FIG. 9.

Fig. 9, thus eliminating the necessity for this figure in most cases. This method would indicate at once the range over which the pump could be used with good results. The application of this is shown in Fig. 10, curve I. With the efficiencies indicated in this way it is seen that if 70 per cent. is the minimum efficiency desired for a pump of this type, the specific speed must be between 4,200 and 5,100.

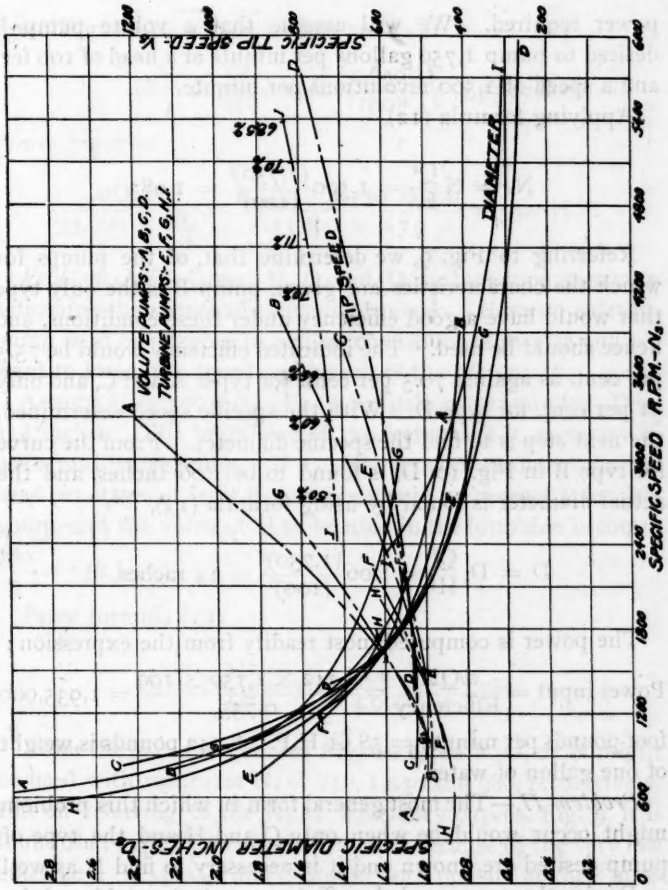


FIG. 10.

APPLICATION TO THE SOLUTION OF PROBLEMS.

Problem I.—To choose a new pump for certain conditions. When a new unit is to be selected certain conditions are fixed by the nature of the problem. A common case would be where the capacity, head and speed are known and it is required to find the size of the pump, its efficiency, and the

power required. We will assume that a volute pump is desired to pump 1,750 gallons per minute at a head of 100 feet and a speed of 1,500 revolutions per minute.

Applying formula (12),

$$N_s = N\frac{Q^{\frac{1}{2}}}{H^{\frac{3}{4}}} = 1{,}500\frac{(1{,}750)^{\frac{1}{2}}}{(100)^{\frac{3}{4}}} = 1{,}983.$$

Referring to Fig. 9, we determine that, of the pumps for which the characteristics are given, pump B is the only type that would have a good efficiency under these conditions, and hence should be used. The indicated efficiency would be 75.5 per cent. as against 70.5 per cent. for types A and C, and only 51 per cent. for type D. With the specific speed determined, the next step is to find the specific diameter. From the curve for type B in Fig. 10, D_s is found to be 1.06 inches, and the actual diameter is found by using formula (13),

$$D = D_s\frac{Q^{\frac{1}{2}}}{H^{\frac{3}{4}}} = 1.06\frac{(1{,}750)^{\frac{1}{2}}}{(100)^{\frac{3}{4}}} = 14 \text{ inches.}$$

The power is computed most readily from the expression:

$$\text{Power input} = \frac{wQH}{\text{Efficiency}} = \frac{8.342 \times 1{,}750 \times 100}{0.755} = 1{,}935{,}000$$

foot-pounds per minute = 58.61 H.P. 8.342 pounds is weight of one gallon of water.

Problem II.—The most general form in which this problem might occur would be when only Q and H and the type of pump desired are known and it is necessary to find N as well as D, also the power and the efficiency. As in problem I, let us assume that Q = 1,500 gallons per minute and H = 100 feet. We will assume that the volute pump is to be type D. From Fig. 11, the best efficiency of 75 per cent. requires a value for N_s of about 3,600. Using formula (12) we have

$$N = N_s\frac{H^{\frac{3}{4}}}{Q^{\frac{1}{2}}} = 3{,}600\frac{(100)^{\frac{3}{4}}}{(1{,}750)^{\frac{1}{2}}} = 2{,}722 \text{ revs. per min.}$$

From Fig. 10 $D_s = 0.66$ inches, and

$$D = D_s \frac{Q^{\frac{1}{2}}}{H^{\frac{3}{4}}} = 0.66 \frac{(1,750)^{\frac{1}{2}}}{(100)^{\frac{3}{4}}} = 8.73 \text{ inches.}$$

Power input =

$$\frac{wQH}{33,000 \times \text{eff.}} = \frac{8.342 \times 1,750 \times 100}{33,000 \times 0.75} = 59.0 \text{ H.P.}$$

Problem III.—Given H, Q and D, turbine-type pump to determine the speed, efficiency and power. Here an existing pump is to be adapted to certain conditions of service and we want to know the speed and power of the motor.

Assume H = 700 feet; Q = 4,000 gallons per minute; D = 31½ inches. The high head of 700 feet makes it necessary to use a multistage pump. If 150 feet is taken as the maximum head per stage, at least five stages would be necessary for this pump, and the value of H to be used in the formulae becomes $\frac{700}{5} = 140$ feet.

From formula (13)

$$D_s = D \frac{H^{\frac{3}{4}}}{Q^{\frac{1}{2}}} = 31.5 \frac{(140)^{\frac{3}{4}}}{(4,000)^{\frac{1}{2}}} = 1.769.$$

From the D_s *vs.* N_s curves Fig. 10 types E, G, or H might be used with values for N_s of 810, 1,050 and 980, respectively. But on referring to the N_s *vs.* Efficiency curves, Fig. 9, it is found that type E is entirely unsuited because of the very low efficiency. Types H and G would both operate at their maximum efficiencies; H at 64.5 per cent. and G at 79 per cent., so that, other things being equal, type G would be chosen, thus fixing the value of N_s at 1,050. From formula (12) we find.

$$N = N_s \frac{H^{\frac{3}{4}}}{Q^{\frac{1}{2}}} = 1,050 \frac{(140)^{\frac{3}{4}}}{(4,000)^{\frac{1}{2}}} = 698.5, \text{ say } 700 \text{ revs. per min.}$$

$$\text{Power input} = \frac{wQH}{33,000 \times \text{Eff.}} \quad \frac{8.342 \times 4,000 \times 700}{33,000 \times 0.79} = 896 \text{ H.P.}$$

Here the total head of 700 feet is used to find input for the pump.

Problem IV.—Given H, N and D, turbine pump, to find Q, efficiency and power. Given H = 70 feet; N = 1,600 revs.; D = 16 inches.

Applying formula (15)

$$V_s = \frac{\pi}{12}\frac{ND}{H^{\frac{1}{2}}} = \frac{\pi}{12}\frac{1{,}600 \times 16}{70^{\frac{1}{2}}} = 800.$$

From Fig. 10 we find that types G and I are the only turbine pumps that will meet the requirements. For G, N_s = 2,535 and for I, N_s = 4,950. Referring to Fig. 9 to find the corresponding efficiencies that may be expected, we find type G (for H_s = 2,535) has an efficiency of only 45.6 per cent., while I (with N_s = 4,950) has an efficiency of 70.3 per cent. Hence, though either type of pump might be used, one would operate much more efficiently than the other. Further, though the pumps are identical in diameter and run at the same speed, they would have widely different capacities. This can be seen by applying formula (12):

$$Q^{\frac{1}{2}} = \frac{N_s H^{\frac{3}{4}}}{N},$$

or

$$Q = \frac{N_s^2 H^{\frac{3}{2}}}{N^2},$$

that is, the capacities of two pumps of the same size running at the same speed and under the same head will vary directly as the squares of the specific speeds.

Hence

$$\frac{Q_1}{Q_6} = \frac{(4{,}950)^2}{(2{,}535)^2} = 3.82,$$

or type I would pump 3.8 times as much water as type G. The actual quantity for type I would be

$$\frac{4{,}950^2 \times 70^{\frac{3}{2}}}{1{,}600^2} = 5{,}610 \text{ gallons per minute.}$$

The horsepower input is found from the expression

$$\text{H.P.} = \frac{wQH}{33{,}000 \times \text{Efficiency}} = \frac{8.342 \times 5{,}610 \times 70}{33{,}000 \times 0.703} = 141.3.$$

CORRECTION OF TEST DATA TO A STANDARD BASIS.

Problem V.—Mention was made of the fact that the fundamental relations between Q, H, N and power furnish a convenient means for reducing observed test results to a uniform basis. During tests it may be difficult to adjust, for instance, the speed to the exact value desired, but the results can easily be reduced to the proper values by the use of the relations given. For example, from a test we have N = 1,690 revolutions per minute, Q = 610 gallons per minute, and H = 101 feet. It is desired to find the proper values of Q and H for a speed of 1,700 revolutions per minute, in order to plot a constant speed characteristic. From equations (1) and (2) we have that

$$Q \propto N \text{ and } H \propto N^2.$$

Hence

$$Q_{1700} = 610 \times \frac{1{,}700}{1{,}690} = 614 \text{ gallons,}$$

and

$$H_{1700} = 101 \frac{(1{,}700)^2}{(1{,}690)^2} = 102.1 \text{ feet.}$$

In a similar way test results may be adjusted when either a constant head or a constant quantity characteristic is desired.

LUBRICANT TESTING.

By Lieutenant Commander G. J. Meyers, U. S. Navy, Member.

The importance of the subject of friction and lubrication is evident to every engineer, as it must be considered in some one of its phases in any piece of machinery that moves. The resulting friction from the movement of one part over another is practically an unknown quantity as far as its relations with the variables that control it are concerned. With cylindrical bearings these variables are:

1. Temperature;
2. Speed of the journal;
3. Load;
4. Diameter of the journal;
5. Length of the bearing;
6. Arc of contact of the bearing.

It is evident that before intelligent investigation of lubrication can be made the relation of friction to each of the above variables must be determined. This relation, reduced to proper formulas, can then be used to reduce all results obtained to one standard or may be used to determine the standard conditions under which all testing should be conducted.

After this determination, what are the characteristics of a lubricant that will tell us whether it is suitable for our purpose? There are three of these characteristics:

1. A lubricant must form a film between the moving parts under a given initial load as soon as possible after motion begins. As lubrication is a function of speed, the film of

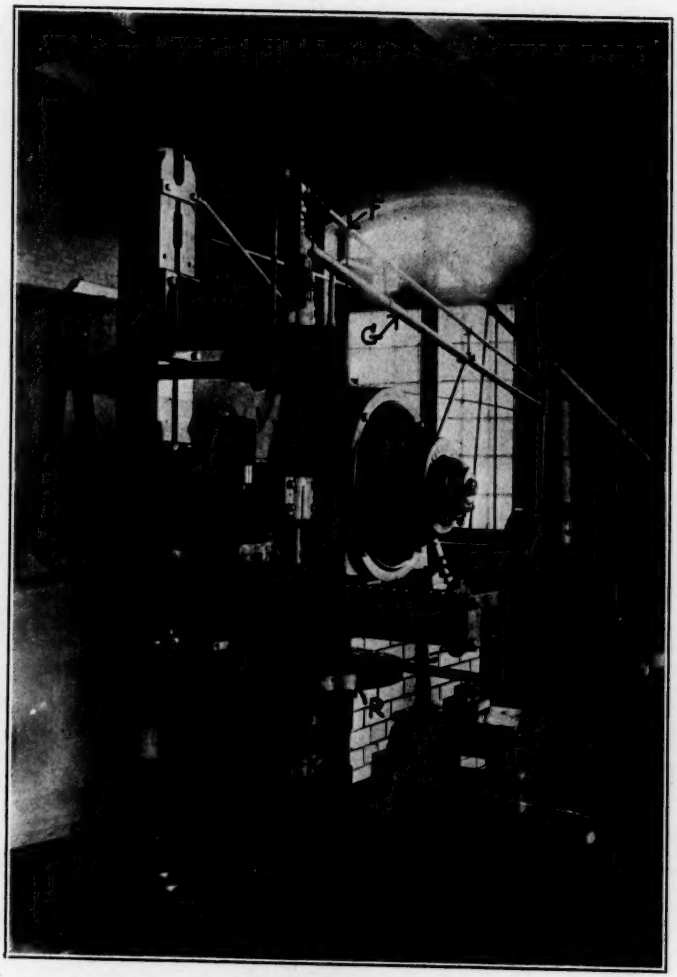


FIG. 1.

lubricant will not form until a certain speed is attained if the bearing is under a certain initial load.

2. A lubricant must maintain the film between the moving parts under the heaviest loads that will be applied.

3. A lubricant should possess ability to be used over and over again; that is, it should have high endurance.

With the object in view of investigating the above properties a machine was built by the author while attached to the Post Graduate School at the U. S. Naval Academy and experimental work was conducted until interrupted by orders to sea duty. The results of this work, while incomplete, are offered as probably of considerable value in any future work along the same lines. A brief description of the machine follows:

General.—A photograph of the machine used is shown in Fig. 1. The journal is screwed to a vertical hollow shaft, V, the journal itself being hollow. The bearings are contained and supported in a box, W, surrounding the journal, a bearing being placed on two opposite sides of this box. This box is oiltight except where the vertical shaft pierces the lid, and contains the oil under test. In this manner the bearing surfaces and the entire journal are surrounded by a bath of oil. The oil may be drawn from the bottom of the oil box, or the drain in the bottom of the oil box may be left open if it is desired to investigate any other than bath lubrication.

Load.—The load is applied by screwing up on A and compressing the triple concentric helical spring B. One end of this spring bears against a plate at the end of rods C, the other ends of these rods attach to a plate, D, which bears against the bearing opposite the spring. The other end of the spring bears against a plate which is in contact with the back of the bearing on the same side as the spring. Compressing the spring results in pressing both bearings against the journal. Loads from about zero up to 15,000 pounds may be applied by using different combinations of these three springs. The amount of load is determined by reading the amount of deflection indicated by the pointer, E, the springs

having been calibrated in a tensile testing machine. The load may be held constant or varied.

Speed.—The speed of the journal is maintained by a 10-H.P. motor. Speed is varied by using the Ward Leonard system of drive. The journal may be driven at speeds anywhere between about 15 revolutions per minute to about 2,000 revolutions per minute. The speed is measured by a Hopkins electrical tachometer.

Temperature.—The temperature is controlled by circulating water, steam, or brine from an ice machine through the inside of the journal and around the outside of the oil box. The temperature-controlling agent flows through pipe F, inside the shaft to the bottom of the shaft, then into the journal, then into the shaft on the outside of pipe F, to the top of the shaft, and discharges through pipe G. It flows to the outside of the oil box from pipe H by means of a flexible rubber tube and discharges through cocks I. The temperature is measured by a thermometer supported by the lid of the oil box. This thermometer is put through a hole in the lid, its bulb being placed at the off-side of one of the bearings at about the middle of the bearing close to both bearing and journal. About a half hour after starting to cool or heat, the journals, bearings, and oil bath are very nearly the same temperature, and the temperature of all will be indicated within a degree by this one thermometer.

The main feature of this machine is the almost absolute temperature control. Unless temperature is controlled in some way the data obtained on any machine are very apt to be misleading. While the temperature of the four variables—bearings, journal, oil film and oil bath—could not be controlled independently, the temperature of all together could be held constant at about the same point. In machines built heretofore the temperature has usually been allowed to run riot, and as oil in its lubricating effects is more sensitive to temperature than to any other factor, the data obtained has never been satisfactory.

Friction.—Friction is measured on a beam scale. With a

7-inch journal the total friction can be measured within 0.237 of a pound. The oil box, bearings and load spring are free to turn, nothing resisting the force of friction between journal and bearings but the friction of the wire bearing, K, which supports the weight of these parts, and the scale. The end of one arm of a bell-crank lever is attached to the plate L at M, the pivot of the bell-crank lever is fixed to the table R at N; the other arm of the bell-crank lever is attached to a wooden shoe resting on the scale pan. When the scale is at a balance the friction of the wire bearing, K, is negligible, and if there is any vibration of the scale this friction tends to dampen the oscillations of the scale beam.

Cleaning Oil-Box Bearings, etc.—The table, R, may be lowered so that the top of the oil box is clear of the bottom of the journal, and may then be swung to one side with the pedestal, P, as a pivot, by disconnecting the bell-crank lever at M and coming up on the bolts that secure table R to the pedestal, P. The bearings may be removed from the oil box, and the oil box removed for cleaning when changing oils. After each test of an oil the bearings, journal and oil box should be cleaned of oil with gasoline after removing all the oil from them.

Pressure of Oil Film.—The pressure of the oil film at any point of the bearing may be determined by ordinary pressure gage attached to the bearing and connected with the oil film by the necessary holes. These holes should be plugged on the outside when the bearing is in use with the gage detached.

Fit of the Bearings.—In order to get regular and consistent results the bearings must be allowed to wear to a fit, as no amount of scraping will result in as perfect a fit as can be obtained by actually running the journal at a moderate speed with a moderate load on the bearing. To expedite the fitting of the bearing in this manner a bar of sapolio was thoroughly dissolved in about a quart of water and to this was added about a quart of oil. This mixture was then used to lubricate the bearings, the machine being run for about sixteen hours.

This process was performed twice and the bearings maintained their fit throughout the tests. The constant running of the machine during the tests improved this fit considerably. In running endurance tests, however, there will undoubtedly be abrasion of the bearings at the end of each run, and the bearings will have to be ground after each one of these tests.

Quantity of Oil Used.—In all tests except endurance tests about five pints of oil are required to cover the journal. For endurance tests the clearance in the oil box is reduced so that a quart of oil can be used. To make all the desired tests of an oil will require about a gallon and a-half of oil.

Size, etc., of Journal and Bearings.—All data recorded in this report were taken with a journal 7 inches in diameter. Each of the two bearings used was 6 inches long and had an arc of bearing on the journal of 90 degrees.

Before starting work of testing, the following list of observations was made up and it was attempted to obtain as many data covering this list as was possible in the three months allowed for the work, but, as the machine was only in its experimental stage and had not been tried out, delays were encountered in making small alterations, adjustments, repairs, etc., that could not be foreseen until after the work had started:

(*a*) Relation between temperature and friction with speed and load constant, varying the temperature.

(*b*) Relation between speed and friction with temperature and load constant, varying the speed.

(*c*) Relation between load and friction with temperature and speed constant, varying the load.

(*d*) Relation between speed, load and temperature and the maximum pressure of the oil film.

(*e*) Comparison between wick-feed and bath lubrication.

(*f*) Determination of the critical speed of several oils, and by comparison to note if there is a wide enough difference in the values to make this one factor in the selection of oils.

(*g*) Endurance test of oils with a given quantity of oil for each test. The temperature to be about 200 degrees F., speed

about 50 feet per second, and load not less than 200 pounds per square inch. Temperature, speed and load to be held constant, the two variable arguments being time and friction.

(*h*) Determination of the film breakdown loads for various oils, and by comparison to note if the values found can be used as a factor in the selection of oils.

(*i*) A test of the various forms of oil grooves found in service and a determination of their relative values.

(*j*) Determination of the value of graphite in oil.

(*k*) Determination of the value of friction for various sized journals with bearings of constant length and constant arc of bearing.

(*l*) Determination of the value of friction for bearings of different lengths and different arcs of contact, with special reference to clearance, chamfer at the edges, relation between radius of the journal and radius of the bearings, and a final determination of the most efficient arc of bearing.

(*m*) Tests of greases.

(*n*) Test of an oil that has all the black matter that ordinarily deposits under heat removed by special treatment.

Data were obtained for (*a*), (*b*), (*d*), (*f*) and (*j*) as follows:

TABLE I.

Oil Number.....	2	2	2	3	3	4
Speed, ft. per sec.....	1.07	1.98	3.36	2.66	3.05	1.86
Load, lbs. per sq. in..	15.1	15.1	15.1	15.1	15.1	32.7
	Friction,					
Temperature, degs. F.	pounds per square inch of bearing surface.					
200	...	.104	.104	.097	.097	.122
184	.093	.107	.111	.101	...	...
141	.104	.122	.143	.129	.101	.172
121	.118	.158	.186	.143	.115	.186
107	.197	.304	.373	.179	.193	.236

Relation between Friction and Temperature.—Table I gives the results of several runs with constant speed and load and with variable temperature of journal, bearings, oil bath and oil film. The results verify the law that friction of lubricated surfaces decreases with rise of temperature. The decrease is

greater at the higher speeds; and as the temperature rises the friction becomes more nearly constant for variable speeds.

TABLE II.

Oil No.	2		5		1		4		5 plus 0.25 per cent. graphite.	
Temp. degs. F.	84		70		72 to 78		84		70 to 76	
Load, lbs. per sq. in.	15.1		32.7		32.7		32.7		32.7	
	Speed.	Friction.	Speed.	Friction.	Speed.	Friction.	Speed.	Friction.	Speed.	Friction.
	.27	.115	1.44	.229	1.44	.208	1.86	.236	1.44	.233
	.52	.140	1.53	.236	1.53	.211	2.35	.244	1.53	.240
	.82	.172	1.89	.251	1.89	.222	2.66	.247	1.89	.256
	1.07	.197	2.35	.269	2.35	.236	3.05	.253	2.35	.276
	1.37	.265	2.75	.286	2.75	.244	3.85	.276	2.75	.290
	1.98	.304	3.39	.304	3.21	.256	4.61	.286	3.39	.312
	2.63	.344	3.66	.315	3.73	.265	...	...	3.64	.319
	3.36	.373	4.12	.333	4.15	.272	...	...	4.15	.333
	4.09	.416	4.37	.348	4.64	.283	...	...	4.43	.344
	4.55	.426	5.86	.361	5.86	.290	...	...	5.80	.391
	...	...	6.23	.365	6.20	.297	...	...	6.23	.398
	...	...	7.24	.394	7.51	.348	...	...	7.24	.416
	..	...	7.64	.398	8.40	.358	...	...	7.63	.426
	...	...	8.39	.416	9.47	.387	...	...	8.40	.437
	...	...	9.28	.441	10.38	.401	...	...	9.28	.448
	...	...	...	...	11.54	.416	...	...	10.69	.477
	...	...	...	...	13.44	.455	...	...	...	...
	...	...	....	...	16.80	.487	...	...	...	...

Speed of journal is in feet per second.
Friction is in pounds per square inch of bearing surface.

Relation between Friction and Speed.—Table II gives the results of runs with constant temperature and constant load with variable speed. The results were plotted and the following formulas were deduced:

Mineral oil No. 2 . . . $r = .10 + .10381v^{.8}$
Mineral oil No. 5 . . . $r = .155 + .05304v^{.8}$
Mineral oil No. 1 . . . $r = .160 + .03725v^{.8}$
Mineral oil No. 4 . . . $r = .192 + .02768v^{.8}$

where r = friction per square inch of bearing surface,
and v = speed of journal in feet per second.

Substituting in these formulas values of V from zero up to the highest values used in each case, curves were plotted and compared with the actual results. The close coincidence of the actual results and the results calculated from the above formulas show that the formula, $r = K + K_1 v^{.8}$, is very nearly correct for speeds above the critical speed, at *ordinary temperatures*. K and K_1 are constants depending on the oil used, K not varying as much as K_1 with rise of temperature.

TABLE III.

Oil No. 6.—Temperature, 74 degs. F.

Load, lbs.pr. sq. in.	15.1		32.7		48.3		67.2		83.9		101.5	
	Speed.	Pressure.	Speed.	Pressure.	Speed.	Pressure.	Speed.	Pressure.	Speed.	Pressure.	Speed.	Pressure.
	1.56	50	2.35	94	2.35	146.5	3.12	185	3.12	235	3.12	290
	2.35	50	3.12	94	3.12	145.5	3.88	185	3.88	235	3.88	290
	3.12	49	3.88	94	3.88	144.5	4.70	185	4.64	235	4.70	290
	3.88	48	4.70	93	4.70	142.0	6.17	185	6.17	235	6.17	285
	4.70	47	6.17	92	6.17	142.0	7.39	183	7.21	235	8.09	280
	5.71	46	7.08	91	7.45	141.0	9.16	181	8.09	235	10.23	278
	6.17	46	8.09	91	8.28	140.0	10.23	180	9.10	234	11.60	277
	7.11	46	9.16	90	9.16	139.0	12.22	178	10.23	234	13.65	276
	7.48	46	10.29	90	10.29	138.0	14.05	177	11.60	233	15.27	275
	8.25	46	11.66	90	11.63	137.5	15.27	176	12.22	232	...	...
	9.16	46	12.28	90	12.31	137.5	...	...	12.64	232	...	...
	10.23	46	13.74	90	13.74	137.0	...	...	13.13	232	...	...
	11.60	45	15.27	90	15.27	136.0	...	...	14.50	230	...	...
	12.28	45	...	...	...	...	...	...	15.27	229	...	...
	13.74	45	...	...	...	...	...	...	...	...	...	...
	15.27	45	...	...	...	...	...	...	...	...	...	...

Speed of journal is in feet per second.

Pressure is the maximum pressure of the oil film in pounds per square inch by gage.

Relation between Speed, Load and Temperature and Maximum Pressure of the Oil Film.—By attaching a pressure gage to one of the bearings at Y, Fig. 2, with a hole drilled as shown to connect with the oil film at X, the maximum pressure of the oil film was recorded by the gage. The first set of data was taken at constant temperature for several different loads, with variable speed for each load. These data

are contained in Table III. The maximum pressure increases with the load and seems to decrease slightly with increase of speed. While speed seems to be necessary to obtain this pressure, it is not necessary to have speed to maintain it, as the pressure will remain until the load is released after stopping the journal.

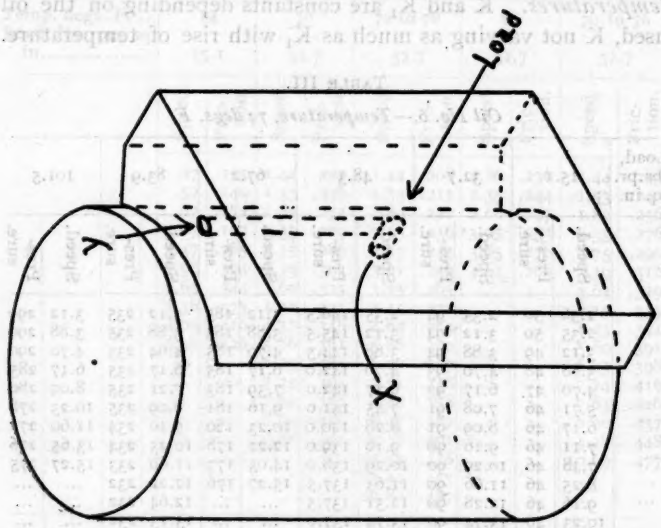


FIG. 2.

The second set of data was taken at four different loads with speed constant for each load but with variable temperature. These data are contained in Table IV. The maximum pressure increases with temperature from about *three* times the load per square inch at ordinary temperature to about *six* times the load per square inch at 200 degrees F.

Table V gives the results for another oil. A comparison of Table V with Table III shows that the maximum pressure of the oil film is also a function of the oil used.

Both Tables III and IV illustrate the folly of attempting to introduce oil to the bearing at a point anywhere near the line of load, and also illustrate the feebleness of twenty pounds

pressure or less on turbine forced-lubrication systems where the oil is introduced very near the bottom of the bearing.

Critical Speed.—After eliminating undesirable oils by suitable chemical tests, it has been attempted to devise some physical test that will be practical in the further elimination of oil below a certain standard.

TABLE IV.

Oil No. 6.

Load, lbs. per sq. in..	15.1		32.7		49.3		60.3	
Speed, ft. per sec.....	2.29		2.29		3.12		2.12	
Temp. °F.	Pressure.	Temp. °F.	Pressure.	Temp. °F.	Pressure.	Temp. °F.	Pressure.
80	51.5	83	97	87	146	81	166
84	52	94	97	90	146	86	166
87	52.5	98	97	98	146	90	166
94	53.5	103	97.5	106	145	94	171
100	54.5	114	103	126	145	98	174
108	56	137	109	147	160	108	177
116	57	147	111	154	167	116	181
126	57.5	154	114	164	172	133	183
136	59	166	116	172	175	138	185
146	60	174	120	192	225	154	190
170	64	183	130	196	235	170	205
180	70	196	140	200	262	192	250
186	73	198	142	...	...	196	270
192	75	200	145	...	...	198	280
200	80	202	147	...	...	200	295
204	87	206	153	...	...	...	...

Temperature is that of journal, bearings, oil film and oil bath.

Pressure is the maximum pressure of the oil film in pounds per square inch by gage.

TABLE V.

Oil No. 2.—Temperature, 76 Degrees F. to 80 Degrees F.

Load, lbs. per sq. in.	15.1		32.7		49.3		67.2		83.9		101.5	
Speed.	Pressure.	Speed.	Pressure.	Speed.	Pressure.	Speed.	Pressure.	Speed.	Pressure.	Speed.	Pressure.
6.17	55	6.17	96	6.17	146	6.17	173	6.17	225	6.17	273
9.16	54	9.16	95.5	9.16	144	9.16	169	9.16	225	9.16	269
12.22	54	12.22	95	12.22	142	12.22	168	12.22	222	12.22	268
15.27	54	14.35	95	14.20	142	13.59	167	15.27	222	15.27	268

Speed of journal is in feet per second.

Pressure is the maximum pressure of the oil film, pounds per square inch by gage.

One of the physical properties of a lubricant is its ability to form an oil film when the journal is under a certain load when the journal reaches a certain speed. The speed at which

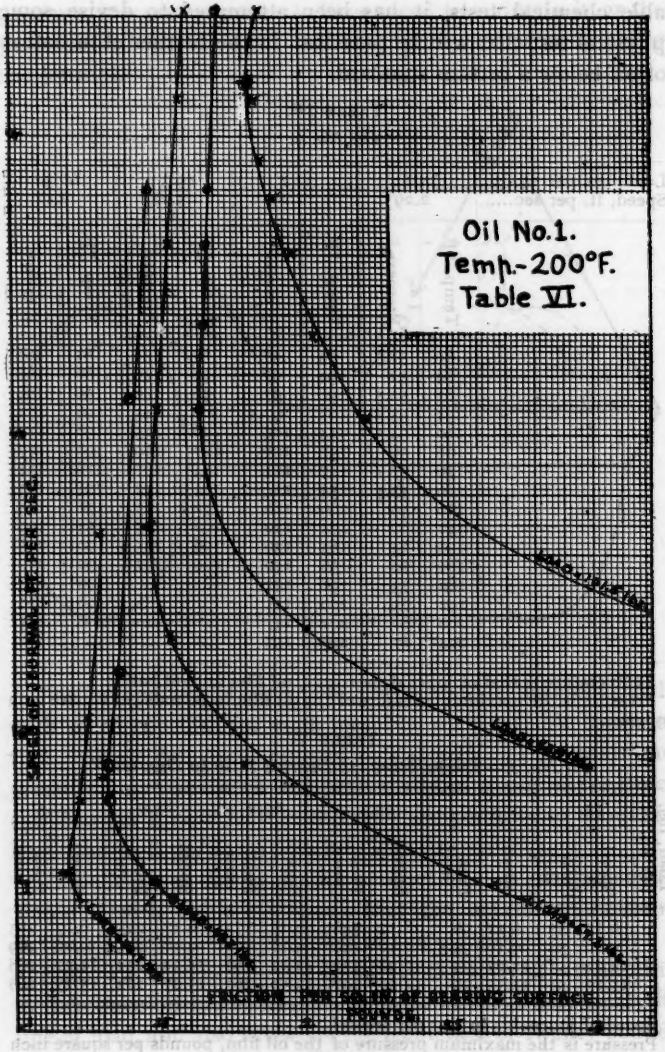


FIG. 3.

the oil film forms under a given load is called the critical speed for that load. As one of the functions of the lubricant is to *form* a film between the two rubbing surfaces and thus reduce friction and wear, and as proper lubrication is directly

TABLE VI.

Oil No. 1.—Temperature, 200 degrees F.

Load, pounds per square inch.	Friction, pounds per square inch of bearing surface.	Speed of journal, feet per second.	Critical speed, feet per second.
32.7	.122	1.44	1.53
	.118	1.53	
	.122	1.77	
	.125	2.05	
	.129	2.66	
49.3	.154	1.44	1.85
	.147	1.50	
	.132	1.77	
	.132	1.89	
	.136	2.20	
	.140	3.12	
	.147	3.82	
67.2	.276	1.44	2.69
	.265	1.50	
	.179	1.89	
	.154	2.32	
	.147	2.69	
	.150	3.08	
	.154	3.63	
83.9	.516	1.44	3.08
	.297	1.89	
	.201	2.35	
	.165	3.08	
	.166	3.36	
	.167	3.63	
			4.18
101.5	.337	2.32	
	.222	3.05	
	.204	3.33	
	.197	3.60	
	.190	3.79	
	.186	3.82	
	.184	4.12	
	.182	4.18	
	.186	4.43	

a function of the speed of the journal, the critical speed should be taken as one of the factors in selecting lubricants. That is, if for each of several different loads the critical speed is found for each lubricant, then, with the loads and cor-

responding critical speeds as arguments, a curve of critical speeds is plotted for each lubricant tested, that lubricant which has the lowest critical speeds should be the best of the lubricants tested. This curve for each lubricant shows the lubricant's ability to *form* the film.

The critical speeds for four different oils were found as follows:

The load and temperature were maintained constant for each test. The journal was first run at a speed below the critical speed, and changed by small increments, the friction being measured and recorded at each speed. Below the critical speed the friction decreased with increase of speed and reached a minimum at the critical speed and then started to increase, following nearly the relation, $r = K + K_1 V^{.8}$. At the point where friction was least the speed was read off and tabulated as the critical speed. This point was found at several different loads. This process is illustrated in Fig. 3, which is typical of each lubricant tested. (Table VI.)

Then with the loads and corresponding critical speeds for each oil as arguments, the curves in Fig. 4 were plotted. These show that oil No. 2 is better than the other oils; that is, it will more readily form a film between the journal and the bearings. The order of merit of these four oils is as follows:

First, ...	Oil No. 2;
Second, ...	Oil No. 3;
Third, ...	Oil No. 4;
Fourth, ...	Oil No. 1.

To the best of my knowledge this is about the order of merit of the oils as proven in use and previous tests, oil No. 2 proving in use that it has no equal. (Table VII.)

Stress must be laid on the following facts:

(*a*) It is impossible to obtain any consistent results without maintaining constant temperature of the journal, bearings, oil bath and oil film.

(*b*) Results will be erratic if the bearings do not fit properly.

It has always been a contention that laboratory tests with a bearing and journal are not reliable because in practice the bearings do not fit as well as in the laboratory tests. That this is not true must be granted when it is considered that bearings in constant use will wear to a perfect fit in a short time, and will remain in that condition if abundantly and properly lubricated, and that a bearing when so lubricated,

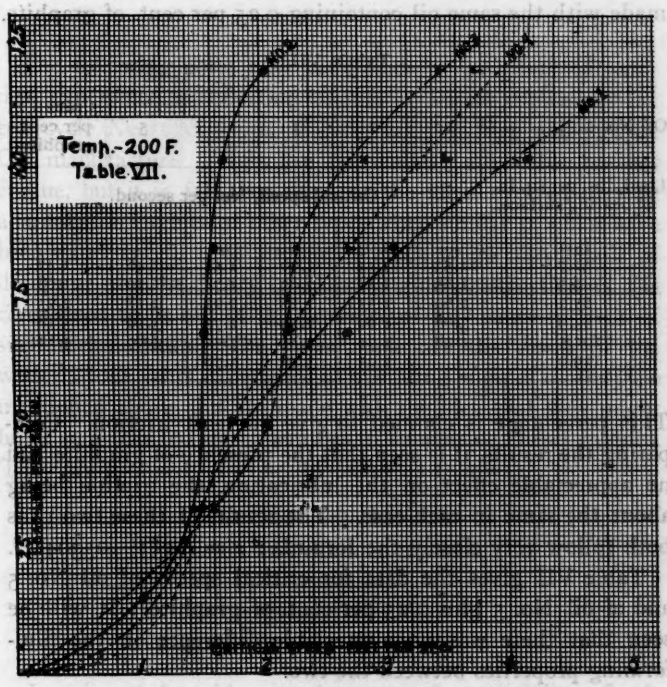


FIG. 4.

if it heats above the normal, is wearing to a perfect fit. Beyond a certain point all the scraping and care that can be given in fitting a bearing will not give it as perfect a fit as the journal itself will give it in actual use. Consequently there is no reason why bearings in service should not be in ideal condition if properly lubricated, and no reason why data

obtained on a laboratory machine should not be perfectly applicable to machinery in service.

Comparison of Mineral Oil with the Same Oil containing 0.25 Per Cent. of Graphite (Table II).—With the load and temperature constant a run was made with mineral oil No. 5, varying the speed and measuring the friction for each speed.

Under as nearly the same conditions as possible a run was made with the same oil containing 0.25 per cent. of graphite.

TABLE VII.

Oil No.....	1	2	3	4	5	5 plus 0.25 per cent. graphite.
Load, lbs. per sq. in. of bearing surface..	Critical speed, feet per second.					
15.1	...	1.03	...	...	...	...
32.7	1.53	1.44	1.60	1.51	1.46	1.47
49.3	1.85	1.51	2.05	1.74	1.79	1.68
67.2	2.69	1.53	2.20	2.21	2.18	2.23
83.9	3.08	1.62	2.29	2.71	3.16	3.18
101.5	4.18	1.71	2.85	3.51	4.09	4.12
119.0	...	2.05	3.46	3.76	...	...

This percentage is claimed to give the least friction. Comparing the results, it is seen that the presence of graphite had no appreciable effect on the lubrication, the friction being about the same in each case. The results of these two runs both follow very closely the formula: $r = .155 + .05304V^{-.8}$.

Table VII gives the data for critical speeds of oil No. 5 and of oil No. 5 plus 0.25 per cent. of graphite. It will be seen that there was no appreciable difference in the film-forming properties between the two.

NOTES ON OPERATION OF SUBMARINE DIESEL ENGINES.

By Lieutenant F. C. Sherman, U. S. N., Member.

The following notes have been made as the result of practical experience in operating submarine Diesel engines of the two-cycle Neurnberg type of New London Ship and Engine Co. manufacture. They are particularly applicable to that engine, but it is felt that some of the ideas evolved may be adaptable to engines of other types when similar troubles have been experienced. No attempt is made to go into details of design or into the theory of Diesel engine construction as it is considered to be out of the province of this paper to treat of such technicalities. The notes made are the result of the writer's four years experience in operating engines of the type mentioned above and are made with the idea that they might be of some use to other operators of submarine Diesel engines. They are grouped into five divisions, as follows:

I. Crank-case explosions.
II. Piston and cylinder troubles.
III. Wrist-pin troubles.
IV. Air-compressor troubles.
V. Auxiliaries.

In spite of the wide spread reports of unreliability of submarine Diesel engines of the past few years, the writer has always maintained that every effort should be made to make the material operate satisfactorily before condemning it as unsatisfactory and unreliable. In many cases further investigation has shown that inexperienced personnel are to blame and not the long suffering and almost always blamed material. Acting under this assumption, every trouble experienced was

thoroughly gone into and, if possible, a remedy for it evolved. The following notes are made from some of the troubles discovered and cured. It is hoped that they will be of some assistance to other Diesel engine operators perhaps wrestling with similar troubles.

I.—CRANK-CASE EXPLOSIONS.

In the type of engine upon which this discussion is based, the crank case is totally enclosed and the oil from the lubricating system, after cooling the piston heads, is drained into the crank case, from which it runs into the settling tank and is used over again. The scavenger air for the working cylinders is compressed in a scavenger cylinder in tandem with the working cylinder and the piston is of the stepped type, the lower step working in the scavenger cylinder and compressing the scavenger air which is forced into a housing around the scavenger cylinder and above the crank case. The scavenger cylinder makes a joint between the crank case and the scavenger housing. Now in every case of crank case explosions it was found that there was a leak from the scavenger housing to the crank case. This joint was packed with soft packing and proved to be very difficult to keep tight, due to the scavenger cylinder holding-down bolts working loose from continual shock. But in every case of crank-case explosion, a leak of scavenger air to the crank case was found, and when it was corrected the explosions stopped. The permanent care taken following this discovery was to test the scavenger housing at frequent intervals for tightness with air pressure and to examine frequently the scavenger cylinder holding-down bolts and to keep them set up tight every time they came loose. These tests were made about once a month, as convenient with the boat's operation, and, following this care, practically all crank-case explosions and their destructive results were eliminated.

As to the theory of why this scavenger-air leak resulted in crank-case explosions it is impossible to give any very satisfactory explanation. The scavenger-air pressure is only about

seven pounds and its temperature from 140 degrees F. to 160 degrees F., although sometimes it would go up to 180 degrees F. It occurred to the writer that the lubricating oil draining into the crank case must be churned up considerably by the cranks revolving therein, which, together with the working heat of the engine, would cause a distillation of the volatile products* of the oil, and form an explosive mixture of vapor* in the crank case, totally enclosed. The ignition point of this mixture must be close to or below the scavenger-air temperature, and consequently when the scavenger air leaked into the crank case the explosion resulted. Whether this is the true explanation or not, it is difficult to say, but it is certain that when there was no leak from scavenger housing to crank case no crank-case explosions occurred.

Another remedy adopted, not as a cure but to reduce the effect of a crank-case explosion, was to install a vent to the crank case. The crank case is totally enclosed only for the purpose of retaining the oil used in the lubricating system and prevent its being splashed about in the engine room. The result was that when a crank-case explosion occurred the gas had no place to escape without wrecking something, blowing off the crank-case doors or housing. To prevent this, a 2½-inch pipe was led from one end of the crank-case housing and left entirely open. The end was taken to the scavenger suction intake for the purpose of furnishing a slight draft to the crank case and removing any gases collected there. This was not entirely necessary, however, as the main purpose of the pipe was to furnish an opening to the crank case to allow the expanded gases of the explosion to get out without blowing something out. As a result of this, if a crank-case explosion occurred, in spite of our efforts to keep the scavenger housing tight, no damage resulted.

Another feature of the leak in the scavenger housing was found to be that the scavenger-housing temperature increased unduly, especially when running at higher powers. This undue

*This explanation seems doubtful. It is more probable that the explosive mixture present consists of oil bubbles filled with hot air.—EDITOR.

rise of temperature probably resulted from slight burning of the oil vapor around the leak causing an increase of the temperature without, however, causing an explosion. A correction of the air leak always resulted in a decrease of scavenger-housing temperatures.

These leaks from the scavenger housing, it must be understood, were not sufficient to cause trouble from drop in scavenger-air pressure and would ordinarily not be expected to give any trouble. The effects on crank-case explosions and housing temperatures, however, were extremely annoying, as outlined above.

Closely allied with the crank-case explosions, but of an entirely different nature, were scavenger-housing explosions. We found these to be due to two causes, namely, presence of a superfluity of oil in scavenger housing and leaky or defective scavenger valves. If there was too much oil in the scavenger air, and the scavenger valve to the working cylinder remained open an instant too long, or leaked after it was closed, the compression temperature or flame from the working cylinder would be transmitted to the scavenger housing and set off the oil and vapor in that chamber, resulting in an explosion liable to wreck the housing, as relief valves fitted there were never efficient in quickly releasing the excessive pressure formed. The remedy for this form of explosion was to keep excessive oil out of the scavenger housing through drains fitted at the bottom and to keep the scavenger valves to the working cylinders functioning properly. With these precautions very little further trouble was experienced with scavenger-housing explosions.

II.—PISTON AND CYLINDER TROUBLES.

Under this head are comprised cracked piston heads, cracked cylinders, cracked pistons and piston seizures. These troubles are almost all traceable to defective cooling of the piston head. On the particular type of engine in question, the pistons were cooled by lubricating oil forced up from the lubricating system through the connecting rod and wrist pin and then up

through a pipe leading to the hollow piston head and thence down on the opposite side through a drain pipe to the crank case, whence it drains by gravity to the settling tanks.

All of the troubles enumerated above were directly attributable to defective cooling of the piston heads. The most frequent cause of this was the presence of salt water in the lubricating oil which remained in the piston head owing to the location of cooling-water inlet and outlet. Due to the temperature to which the oil was subjected, the water would quickly evaporate, leaving a salt deposit in the oil which was black in color, giving the appearance of carbon. For a long time this deposit was thought to be carbon, on account of its color, but an analysis showed it to be over 90 per cent. salt. This salt would form a black, gummy mass, and would soon collect in the piston heads and pipes leading to the piston heads and result in decreasing or blocking altogether the supply of cooling oil to the piston heads. Instantaneously the piston head would get hot and either crack or seize, or heat would travel to the cylinder or piston itself, resulting in cracking the cylinder with its cooling water outside or the piston seizing and cracking. Obviously the remedy for this is to keep salt water out of the oil. However, with an engine using salt water for its cylinder-cooling and oil-cooling medium, this is not as easy to do as it sounds; but it can be done if proper care is exercised. On the vessel on which the writer served it was never completely accomplished until the circulating-water pumps were removed from over the crank case where water leaking slightly past the plungers and stuffing boxes found its way into the crank case and there mixed with the oil. Another frequent source of trouble were the oil coolers, where the oil is cooled by circulating water before being again used in the engine system. The oil passed through nests of tubes surrounded by cooling water, and trouble was experienced in preventing tubes from pitting through and gaskets from leaking. Whereas the oil pressure when the engine was running was greater than the water pressure the leak would become effective when the engine was shut down and the lubricating-oil

pumps stopped. Water would then leak in and cause trouble on the next run of the engine. Another source of leaks was from slight cracks in cylinders, sometimes quite imperceptible to the eye when cylinder was cold, but allowing slight leaks of cooling water to crank case when warm. All of these leaks, wherever they may be, must be prevented to insure proper cooling of piston heads and to prevent troubles ensuing from this source.

III.—WRIST-PIN TROUBLES.

Wrist-pin troubles in brief are due to insufficient lubrication, insufficient clearance, undue wear on bushing or pin, and heating resulting from hot piston or piston head.

The trouble due to insufficient lubrication causing hot wrist pins is in some cases traceable to the salt water in the oil, as explained above, and is caused by clogging of the oil passages as described. The remedy is the same as was there given. In other cases, however, it may be due to improper grooving of the wrist-pin bushing or bearing surface. This subject must be studied in connection with the approved forms of oil grooves for bearings, and steps taken to insure that the oil is being properly distributed on the wrist-pin bearing. As it has no special features, it is not gone into further here. On engines with forced-lubrication pumps operating from the main engine shaft, hot wrist pins frequently develop on first starting up an engine. This is probably caused by lack of lubricating oil in the wrist-pin bearings, the highest part of the system, when the engine is first turned over. This can be prevented on engines with an independent lubricating pump by starting up the auxiliary lubricating pump several minutes before attempting to turn over the main engines and running it long enough to insure getting lubricating oil to all parts of the system. This should always be done before starting up, as frequently wrist pins will run hot and wipe in the few minutes before oil from the attached pumps can get to them.

Insufficient clearance on wrist-pin bushings sometimes results in not allowing sufficient lubricating oil to form a good film on the bearing and causes wiping or heating of the wrist

pin. Good practice is to allow about .002 inch vertical clearance between the pin and bushing and about .006 to .008 clearance on the sides. This additional side clearance gives no more play in the bearing, as the pressure is always vertical, but gives the oil a better chance to circulate in the bearing and form the "oil film" for lubrication.

Undue wear on wrist-pin bushings results in loss of lubricating oil from the bearing due to leakage, and also in loss of compression in the cylinders from the dropping down of the pistons. Good lubrication in a forced-feed system requires reduction of all oil leaks to a minimum, especially where the oil is also required for cooling of piston heads. Every oil leak results in a loss of pressure and thus affects the quantity of oil going through the piston heads for cooling and also the effectiveness of the lubrication through reduction of the oil pressure on each bearing. Consequently anything which can be done to prevent undue wear on the wrist-pin bushings is important. In addition, loss of compression in the cylinders causes inefficient combustion of the fuel, reducing the economy of the engine, and necessitates frequent overhaul and insertion of liners under connecting rods to increase compression or frequent renewal of wrist-pin bushings. This reduces the availability of the engine and is extremely undesirable. This undue wear on bushings may be due to insufficient lubrication on starting up, as described above, and is remedied as there given. However, it may be caused by improperly made wrist pins or bushings. The bushings should be of phosphor bronze, of as tough and durable a composition as possible. The wrist pins are of steel, hardened on their wearing surface by either the bone or cyanide processes. The wrist pins furnished us originally were bored out from one end only and that end plugged with a threaded brass plug. We found that the pins would take a more uniform heat and better hardening if the pins before hardening were bored clear through and both ends plugged with the threaded brass plugs. This was a slightly more costly process, but resulted in much better pins, and is recommended for all wrist pins for Diesel engines. These

steps are about all that can be taken to reduce undue wear on wrist pins and bushings.

Heating of wrist pins from hot pistons and piston heads is caused by heat transference from the above parts and can only be remedied by correcting the cause as discussed under II above.

IV.—AIR-COMPRESSOR TROUBLES.

Diesel engine air-compressor troubles comprise valve trouble, cooler leaks and explosions. They are due to the high temperatures created when the air is compressed in two or more stages from atmospheric pressure to approximately 1,000 pounds per square inch. In the type of engine mentioned at the beginning of this paper, the compressor was designed to take its suction from the scavenger-air housing, and it was then compressed in two stages in tandem to 800 to 1,000 pounds per square inch. This air was cooled from each stage in a cooler consisting of nests of small, straight tubes around which circulated cooling water. The air from the 2d-stage cooler passes to the spray-air bottle which acts as a reservoir on the way to the spray-air line of the engine. It will be seen that when the two-stage air compressor takes its suction from the scavenger housing containing air at seven pounds pressure (above atmosphere) it virtually makes a three-stage compression. On our engine, however, there was always so much oil in the scavenger housing that it was considered dangerous to compress air containing so much oil and subject it to the temperatures reached, and in practice it gave considerable trouble. So the suction to the scavenger housing was disconnected and a suction direct to the atmosphere substituted, which gave a straight two-stage compression from atmospheric pressure to 1,000 pounds per square inch. This worked much more satisfactorily as regards presence of oily vapor in the compressed air and occurrence of cooler explosions.

A common practice in Diesel engine design seems to be to have a restriction in the spray-air line between the reservoir

and the engine. The only object of this that I have been able to discover is to enable a higher pressure to be carried in the reservoir than is needed on the spray-air system, so as to build up a reserve for starting after the engine has been shut down. If this is its purpose it never was successful for us, and only resulted in reducing the amount of spray air we were able to get through to the fuel valves. Furthermore, it would frequently clog up and catch dirt and oil to further reduce the opening, so that in general it was more of a nuisance than anything else. Acting on this belief, the restriction on the spray air was removed entirely and much better results in every way were obtained. Whereas before, poor fuel combustion had been obtained when carrying 800 to 900 pounds pressure on the spray air, after removing the restriction perfect combustion was obtained with as low as 550 to 600 pounds pressure on the spray air. Needless to say, the restriction was never again replaced. This may not be of any advantage on other engines, as the type of fuel valve and size of spray nozzle may not permit of any difference, but for us it worked wonderfully well, giving better fuel pulverization and relieved the air compressor enormously of extra load and correspondingly higher temperatures due to the higher pressure required with the restriction in place.

Another point in regard to air-compressor trouble is cylinder lubrication. The principal danger of this is too much lubrication, allowing oil to be carried into the compressed air and causing high temperatures or explosions from burning or combustion of the oil vapor. The best practice is to eliminate direct cylinder lubrication entirely and depend on the moisture and oily vapor in the engine-room atmosphere to furnish sufficient lubrication. In practice this worked very well for us, and we had no trouble from lubrication while using no oil whatsoever directly on the air compressors.

The principal valve trouble which we experienced was due to the valve springs losing their temper after a few hours' running due to the high temperatures of the uncooled air to which they were subjected. The second-stage suction valve was

the principal source of trouble, and when its spring gave out it would leak, allowing second-stage pressure to back up in the first-stage receiver and increase the work on the first stage and, in general, raise hob. Another source of valve trouble was the gradual collection of carbon deposits on the valve seats due to the presence of oil in the air and causing the valves to leak as described above. Cutting off the oil used for cylinder lubrication helped both the above troubles. But the greatest assistance to correct these faults was a water cup installed on the first-stage air suction and set to feed a small quantity of fresh water into the compressor with the air. This water cup was simply a large oil cup arranged for drop feed, filled with fresh water instead of oil. A fairly rapid feed was set on it, about two to four drops per second, and this water was dropped through the top of the air-suction pipe and drawn into the compressor with the air. The action of this fresh water was found to be as follows: The water helped to lubricate the valves and cylinder walls and prevented the deposit of carbon. The high temperature almost immediately turned it into steam, absorbing some of the heat without rise of temperature in the form of latent heat, and thus keeping down the temperatures developed due to compression. In addition the steam kept the carbon from collecting and gumming up the valves, and the reduced temperatures resulting prevented the springs from losing their temper. This fresh-water cup was a fine thing, and I strongly advise other Diesel engine operators to try it on their air compressors.

Cooler leaks were probably caused by high temperatures and possibly some electrolytic action on the tubes. The installation of the water cups kept down the temperatures and also kept carbon and oil from collecting in the coolers and restricting the heat transference. To prevent electrolytic action the outside of the copper tubes was tinned and small zincs were placed in the cooler. These precautions eliminated almost all of our air-compressor troubles. In addition all clearances were kept down to a minimum, about 1-64 inch on both stages.

V.—AUXILIARIES.

Under this head is included the lubricating-oil pumps, the fuel-oil pumps comprising supply and measuring pumps, and the circulating-water pumps. The principal troubles experienced with auxiliaries were with those geared to the main shaft. These pumps were the reciprocating type and comprised a fuel-feed pump, a lubricating-oil pump and a circulating-water pump, all driven by one large crosshead operated by a crank shaft geared to the main shaft. The first trouble experienced was with the fuel-oil supply pump, which leaked, in spite of efforts to keep it tight, a small amount of fuel oil into the crank case. After mixing with the lubricating oil this caused rapid deterioration of the lubricating oil for lubricating purposes, as well as scavenger and crank-case explosions from its low flash point and volatility. To obviate this trouble the fuel-pump was removed and a gravity fuel feed substituted, the gravity fuel tank being supplied by a small rotary pump, motor driven, secured to the bulkhead. This removed all possibility of getting fuel oil into the lubricating oil and prevented recurrence of its evils. The lubricating oil and circulating-water pumps were the next to give trouble. They were high-speed reciprocating pumps, and it was almost impossible to keep salt water from leaking from the circulating-water pump and finding its way to the crank case and, ultimately mixing with the lubricating oil, causing all the troubles enumerated under division II. In addition, mechanical difficulties with these reciprocating pumps caused by momentary high pressures when the pump became air bound or through defective valve action resulted in frequent stripping of gears and breaking of pump crank shafts. These breakdowns finally became so frequent that independent motor-driven rotary-type pumps were installed for both lubricating oil and circulating water. The reciprocating-pump connecting rods were disconnected and lashed clear, and the salt water connections were blank flanged to prevent any possibility of salt water leaks, and these pumps were not used but were kept available for connecting up in case of failure of the independent pumps. In nearly two years of operating they were never

needed to be used, the independent pumps operating entirely satisfactorily. As a result of this, we became strong advocates of independent auxiliaries for submarine Diesel engines, more especially on account of the power of the main storage batteries being always available and the replenishment of the power being feasible while the main engines were running by floating the main motors on the batteries and pulling enough current to compensate for that used by the auxiliaries.

The fuel-measuring pumps were of the plunger type, driven by an eccentric off the main shaft. The regulation was by means of the suction valves being held open for a part of the discharge stroke to control the quantity of fuel oil discharged to the fuel valves. This regulation was made by the operator through a hand wheel controlling the rocker arm operating the valves. These pumps gave very little trouble except from wear, and it was necessary to renew the plungers and barrels about every six months to a year on this account. The plungers were required to fit the barrels very closely on account of loss of pressure through leakage past the plungers. Aside from this amount of wear these pumps gave no trouble, and the method of control was entirely satisfactory.

It may be only just to conclude these remarks by a statement of the results obtained when paying attention to the details enumerated above. Two engines of the two-cycle type mentioned were operated for a period of nearly two years after the above methods were put into effect, and during this time no major casualty of any kind was experienced. By major casualty is meant one requiring the services of a Navy Yard or tender to repair. No minor casualty requiring more than a few minutes stop to repair occurred, and no cracked piston heads and but one cracked cylinder were experienced during this time. In fact the reliability of these engines was considered excellent by the operating force, and the cruising radius was regarded as only limited by the fuel-oil storage capacity. If Diesel engines will fulfill this requirement no more can be expected of them. In conclusion, if these notes are of any assistance to other operators of submarine Diesel engines their purpose will have been fulfilled.

TEST OF SCHUTTE AND KOERTING EVAPORATOR.

By M. C. Stuart, Associate.

A recent test made at the U. S. Naval Engineering Experiment Station upon a Schutte and Koerting No. 10 Navy Type Evaporator developed some interesting data regarding evaporator performance which may be of value to those concerned with the design or operation of evaporators of this or similar types.

The general design of the evaporator is shown in Fig. 1. The cast-iron shell is fitted with a door which serves as the steam manifold, and into which are expanded the U-shaped copper tubes. The divisions of the manifold are such that the steam makes four series passes through the top five rows of tubes, and the condensate from the top five rows makes four passes through the bottom row of tubes. The door has provision for eight horizontal rows of tubes, but the upper two rows were omitted by the manufacturer and tests were made with six rows and with five rows of tubes installed. The baffling arrangement consists of the annular ring and the hood shown.

The test had for its principal objects the determination of the following results:

(*a*) The capacity which could be obtained when operating with atmospheric vapor pressure and a maximum chlorine content of 2 grains per gallon in the fresh water produced.

(*b*) The effects of water level, steam pressure, vapor pressure and temperature difference upon capacity and chlorine content.

(*c*) The effectiveness of the baffles provided.

(*d*) The effect of reducing the amount of evaporating surface.

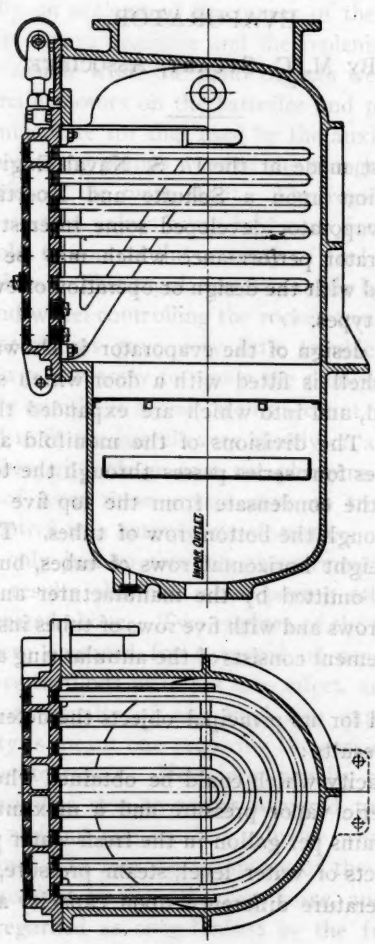


FIG. 1.—CROSS SECTION SCHUTTE KOERTING NO. 10, NAVY TYPE EVAPORATOR.

METHOD OF TEST.

The test consisted of a number of runs of approximately six hours duration, during each of which the vapor pressure, salinity, and water level were maintained constant, and the steam pressure was varied, as will be explained later. The vapor rose through a 3½-inch riser to a distiller condenser located 15 feet above the evaporator. Atmospheric pressure was carried on the distiller condenser, and the condensed vapor flowed by gravity to tanks where it was weighed and its chlorine content determined with the standard water-testing set. The vapor pressure in the evaporator was measured with a mercury manometer and the vapor temperature was measured with a thermometer in a well in the top of the shell. The water level in the shell was observed by means of a gage glass, the upper end of which was connected into the top of the shell rather than in the connection provided, which is just above the top row of tubes. In the usual location of the gage glass, the level in the glass at times fluctuated considerably due to brine splashing through the top connection of the glass, but with the top of the glass connected higher in the shell this fluctuation was absent, and the water in the glass maintained a much steadier level, materially assisting in maintaining steady running conditions.

The steam pressure was controlled by hand, and the temperature pressure and quality of the steam were measured at the inlet to the evaporator. The condensate from the coils was allowed to fall into a drain pot, equipped with a glass, in which a water level was maintained at all times. Air was relieved from the steam space by means of pet cocks in the manifold and drain pot. The relief of accumulated air was found, as usual, to be a necessity for good operation. The condensed steam was cooled and weighed.

The following procedure was carried out in making each run. In the morning the evaporator was partly filled with brine of about $\frac{3}{32}$ds salinity, saved from a previous day's running. Steam was turned on slowly and the evaporator warmed

up with a low steam pressure. The water was brought to a desired level in the glass and maintained at that level throughout the day. Severn River water which has a salinity of about $\frac{1}{4}$ of $\frac{1}{32}$d was used as feed. The salinity in the shell was maintained constant at $\frac{3}{32}$ds throughout the day by the use of continuous or semi-continuous blow-down. No difficulty was experienced in keeping the salinity constant by permitting just enough blow to allow the salinometer to keep floating in the salinometer pot at $\frac{3}{32}$ds. The vapor valve was left open wide and, as atmospheric pressure was carried in the distiller condenser, the vapor pressure in the evaporator was atmospheric or only slightly more than atmospheric. The steam pressure was increased very slowly in regular increments of from 2 to 5 pounds per ten minutes. All readings, including steam and vapor pressures and temperatures, condensed steam and vapor weights, and chlorine content, were taken at intervals of ten minutes throughout the day. The steam pressure was increased until the vapor contained about 2 grains of chlorine per gallon. The steam pressure was then held constant, and a run made for a period of one hour or longer with all conditions constant. The steam pressure was then further increased slowly until the chlorine content reached 5 or 6 grains per gallon, when the steam pressure was again held constant for an extended period. In some cases the pressure was still further increased and in other cases the pressure was reduced and a check run made under the conditions which produced 2 grains of chlorine per gallon.

At the close of each day's run the brine was blown out into a tank in order that it should be available for the start of next day's run. With steam pressure on the tubes, cold water was rapidly pumped into the shell. This resulted in removing a large part of the scale and sludge from the tubes. This water was blown out and the evaporator again filled with Severn River water which was allowed to remain in the shell over night.

DESCRIPTION OF RUNS.

The first series of runs was made with six rows of tubes installed, atmospheric vapor pressure on the shell, and water levels of from 4 to 14 inches above the bottom of the bottom row of tubes. The 14-inch level corresponds to the top of the top row of tubes, but it cannot be said that a water level in the glass at a given level represents an equal submergence of the evaporating surface. In fact, it is generally recognized that the level in the shell of any evaporator is higher than the level as indicated by the gage glass, the difference between the actual level in the shell and the observed level in the glass depending upon the violence of ebullition and possibly other factors, such as salinity and vapor pressure. Nevertheless, the water level as observed in the glass may be used as one of the most important guides to correct operation, and the principal object of the first group of runs was to determine the best level at which to operate, considering the features of maximum capacity, chlorine content and steadiness of operation.

The results of each day's run were first plotted in the form of a log, of which Fig. 2, for run No. 12, is an example. The log shows the increase in steam pressure, temperature difference, capacity and chlorine content throughout the day. The extended periods of one hour or more during which the steam pressures are held constant are designated as runs No. 12*a* and 12*b*, and these portions were used in computing the results tabulated in the table of results given in the insert, Plate I. The log of Fig. 2 shows that certain relations exist between temperature difference, capacity and chlorine content. The relation between temperature difference and capacity need not be shown graphically, for it may be given by a statement of the value of the heat transfer coefficient.

From the standpoint of operation, a most important relation is that between capacity and chlorine content. The operator is vitally interested in knowing the capacity which may be produced without exceeding the limiting value of the chlorine content. Representative curves showing the relation

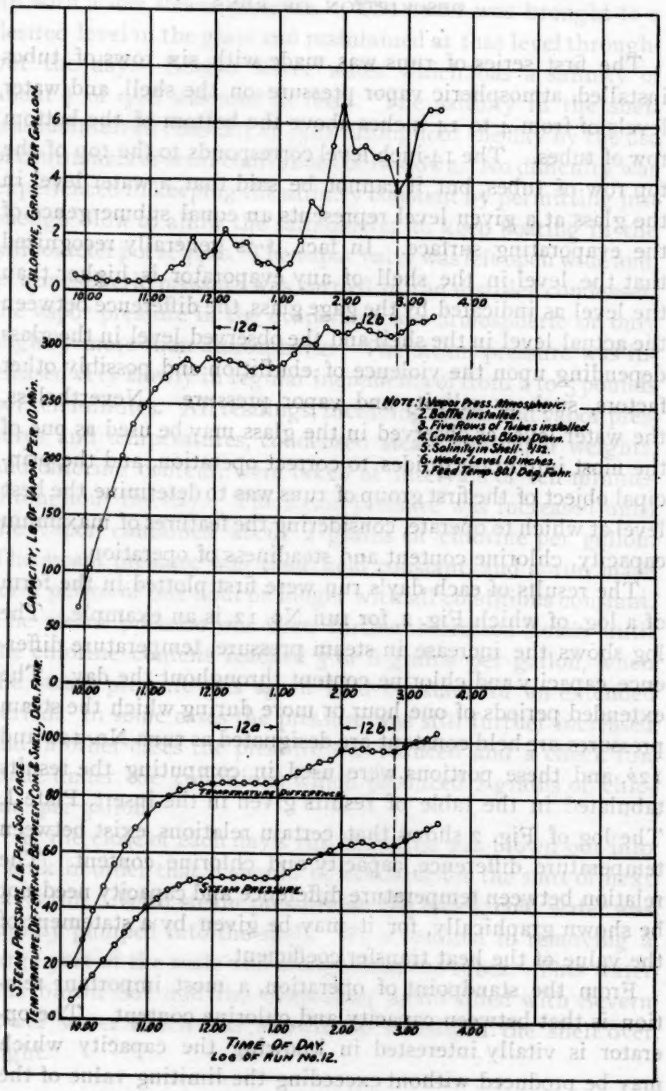

Chlorine, Grains Per Gallon.
Capacity, Lb. Of Vapor Per 10 Min.
Steam Pressure, Lb. Per Sq. In. Gage
Temperature Difference Between Con. & Shell, Deg. Fahr.
12a
12b
Note: 1. Vapor Press. Atmospheric.
2. Baffle Installed.
3. Five Rows of Tubes installed.
4. Continuous Blow Down.
5. Salinity in Shell: 4/32.
6. Water Level 10 inches.
7. Feed Temp. 80.1 Deg. Fahr.
Temperature Difference
Steam Pressure
Time Of Day.
Log of Run No. 12.

FIG. 2.

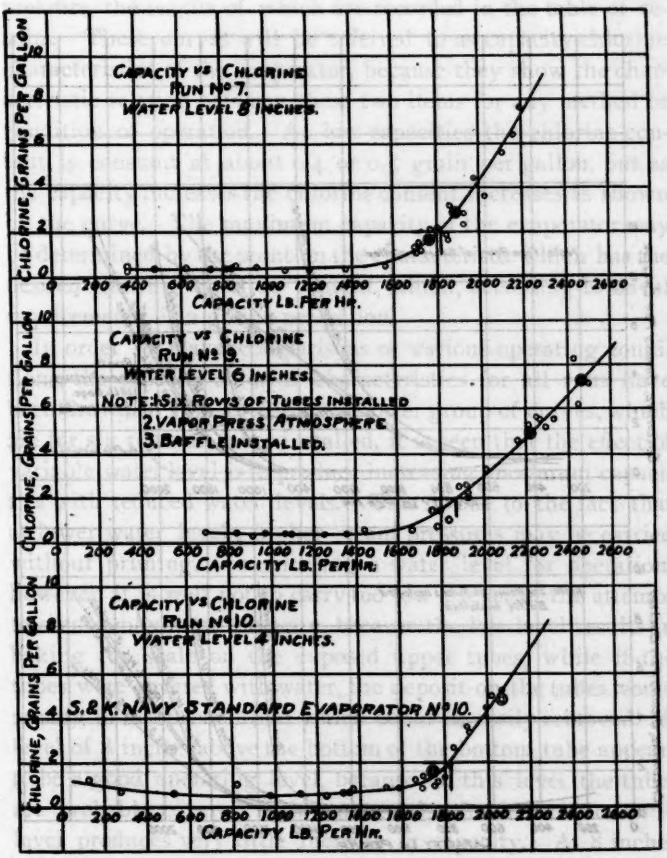

CAPACITY VS CHLORINE
RUN No 7.
WATER LEVEL 8 INCHES.
CHLORINE, GRAINS PER GALLON
CAPACITY LB. PER HR.
CAPACITY VS CHLORINE
RUN No 9.
WATER LEVEL 6 INCHES.
NOTE: SIX ROWS OF TUBES INSTALLED
2. VAPOR PRESS. ATMOSPHERE
3. BAFFLE INSTALLED.
CHLORINE, GRAINS PER GALLON
CAPACITY LB. PER HR.
CAPACITY VS CHLORINE
RUN No 10.
WATER LEVEL 4 INCHES.
S. & K. NAVY STANDARD EVAPORATOR No 10.
CHLORINE, GRAINS PER GALLON
CAPACITY LB. PER HR.

FIG. 3.

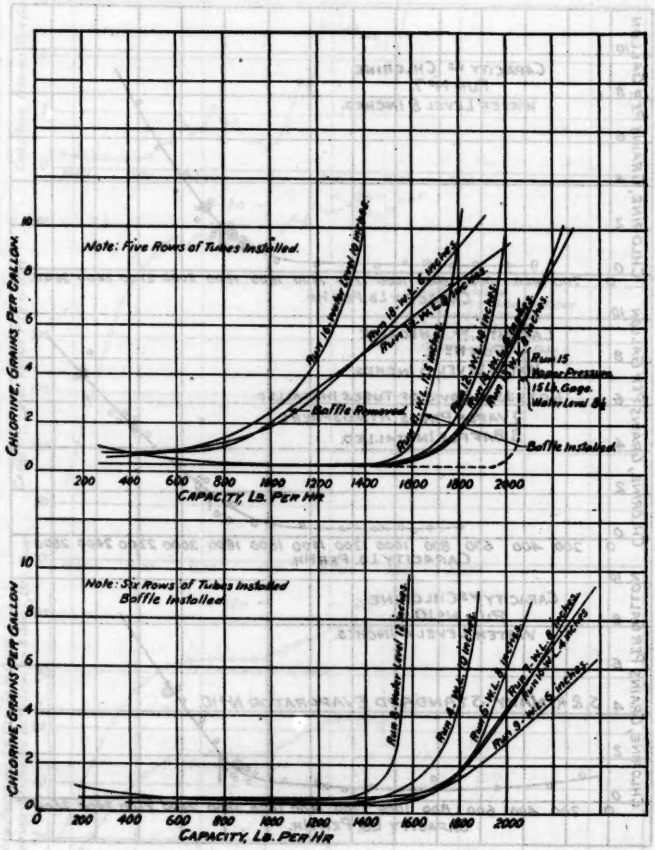

Note: Five Rows of Tubes Installed.
Run 15. Water Level 18 inches.
Run 14. W.L. 6 inches
Run 13. W.L. 3 inches
Run 12. W.L. 2 inches
Run 11. W.L. 1.5 inches
Run 10. W.L. 1 inch
Run 15 Water Pressure 15 Lb. Gage. Water Level 84
Baffle Removed
Baffle Installed
CHLORINE, GRAINS PER GALLON
CAPACITY, LB. PER HR
Note: Six Rows of Tubes Installed Baffle Installed
Run 9. Water Level 12 inches.
Run 8. Water 8 inches
Run 7. W.L. 6 inches
Run 6. W.L. 4 inches
Run 5. W.L. 3 inches
Run 4. W.L. 2 inches
Run 3. W.L. 1.5 inches
Run 2. W.L. 1 inch
Run 1. W.L. 6 inches
CHLORINE, GRAINS PER GALLON
CAPACITY, LB. PER HR

FIG. 4.

between capacity and chlorine content for several of the runs are given in Fig. 3. The small circles represent observations for ten-minute periods, and the larger double circles represent the average values for the extended runs with constant steam pressure, the results of which are recorded in the table of results. These curves will be referred to as capacity-chlorine characteristics of the evaporator, because they show the characteristic relation between these two items for any method or condition of operation. At low capacities the chlorine content is constant at about 0.4 or 0.5 grain per gallon, but as the capacity increases the chlorine content increases as shown by the curve. The maximum capacity of the evaporator may be determined by the point on the characteristic which has the desired limiting chlorine content, which, according to naval requirements, is 2 grains per gallon.

In order to make comparisons of various operating conditions, the capacity-chlorine characteristics for all runs have been drawn in Fig. 4. From the lower group of curves, which are for six rows of tubes installed, it is seen that the effect of variable water level is to produce increasing maximum capacities with reduced water levels. This is due to the fact that at lower water levels, higher steam pressures may be carried without priming. In selecting a water level for operation, however, it is well not to carry too low a level in the attempt to secure maximum capacity, because the low level results in baking the scale on the exposed upper tubes, while if the tubes were covered with water, the deposit on the tubes would remain in a soft condition which could be easily removed. A level of 8 inches above the bottom of the bottom tube appears to be a good operating level, because at this level the tubes are probably covered with water, and a further reduction in level produces very little increase in capacity. At 8 inches level a maximum capacity of 1,825 pounds of vapor per hour is produced. This is equivalent to 21.9 short tons or 5,252 gallons per day. This capacity, with a limiting chlorine content of 2 grains per gallon, could be obtained only by careful operation. In order to establish a safe operating capacity a

capacity may be selected which will permit a certain over load to be obtained. A rated capacity of 1,300 pounds per hour or 3,740 gallons per day would allow a 40 per cent. over load to be obtained if necessary. The chlorine content of the vapor would be 0.4 to 0.6 grain per gallon. The operation would be steady and safe, and would allow a considerable increase in water level or steam pressure without danger of priming.

EFFECT OF AMOUNT OF SURFACE.

In order to determine the effect of the amount of evaporating surface the tubes of the top row were cut out and the holes plugged, and runs made under the same conditions as those with six rows of tubes. The capacity-chlorine characteristics for these runs, Nos. 11, 12, 13 and 14, given in the upper part Fig. 4, are seen to be almost identical with the characteristics for 6 rows of tubes. With 8 inches water level the maximum capacity with five rows of tubes is 1,830 pounds per hour, practically the same as was obtained with six rows of tubes. This appears to prove conclusively what has often been stated, namely, that the maximum capacity which may be produced by an evaporator without excessive chlorine in the vapor is independent of the amount of surface installed, within certain limits, but depends upon such factors as disengaging surface, method of baffling and vapor pressure. However, high temperature differences and higher steam pressures are required to produce a given capacity with the smaller amount of surface. Thus, for a value of heat transfer coefficient, $k = 900$, the maximum capacity of 1,825 pounds per hour will require a steam pressure of 48 pounds gage for six rows of tubes, and a pressure of 66 pounds gage for five rows of tubes. This assumes the same heat transfer factor with each amount of surface. This was practically the case, as may be seen from the table of results.

EFFECT OF HIGH VAPOR PRESSURE.

All of the runs referred to previously were made with atmospheric vapor pressure. In order to determine the effect

of a higher vapor pressure run No. 15 wasmade at 15 pounds gage vapor pressure. The capacity-chlorine characteristic shown dotted on Fig. 4 shows a much higher maximum capacity than was obtained with atmospheric vapor pressure. This is doubtlessly due to the higher vapor density and consequent lower vapor velocity.

However, the higher vapor pressure is not recommended for single-effect operation, on account of the greater ease of operation with atmospheric pressure on both distiller and evaporator and the probability of more difficulty with scale and corrosion at the higher pressures.

EFFECTIVENESS OF BAFFLES.

In order to determine the effectiveness of the baffles the baffles were removed, and runs made at three water levels, with five rows of tubes installed. The capacity-chlorine characteristics of these runs, Nos. 16, 17 and 18, plotted on Fig. 4, show an enormous decrease in capacity obtainable without priming, when compared with the corresponding runs, Nos. 12, 13 and 14 at the same water levels and with the baffles installed. The difference in the shape of the characteristics with and without baffles is also interesting. At 8 inches water level the maximum capacity obtained with two grains of chlorine per gallon is 1,000 pounds per hour, compared to 1,830 pounds per hour with the baffles. This increase in capacity of 83 per cent. establishes the certain effectiveness of the baffles.

HEAT TRANSFER COEFFICIENT.

As has been intimated, the heat transfer coefficient shows the relation between capacity, surface and temperature difference. The average value of K for six rows of tubes installed was 983 and for five rows of tubes was 973. The value of K being practically constant, the product of surface and temperature difference is constant, and the smaller amount of surface requires a higher steam pressure for the same capacity. However, it may be desirable to install the larger amount of sur-

face and operate at a lower pressure, on account of the reduced liability to forming scale at lower temperatures. There is no apparent relation between water level and the value of K, except for extreme low levels, as in the case of Run No. 10, 4-inch level, where K was 822. This low value of K indicates that part of the tube surface was not covered with water.

OPERATION.

From the results of the test the following may be said regarding operation of the evaporator. For best results the water level should be carried at from 8 to 10 inches above the bottom of the bottom row of tubes. As in all evaporator operation, the steam pressure should be slowly increased to the desired value. With six rows of tubes installed and atmospheric vapor pressure, a steam pressure of from 25 to 30 pounds will give the rated capacity of about 1,300 pounds per hour with a chlorine content of less than one grain per gallon. Air should be vented from the steam manifold and drain pot.

Continuous blow-down should be used, and the salinity of the blow-down kept as near $\frac{3}{32}$ds as possible. No difficulty was experienced during the test in operating with continuous blow-down. It is sometimes stated that the heat losses with continuous blow are greater than with intermittent blow, but it has been proven that theoretically the blow-down losses with continuous blow at $\frac{3}{32}$ds salinity equal the losses with intermittent blow-down and a maximum salinity of $\frac{3}{32}$ds.* The fact that the total heat losses during this test were only 3 or 4 per cent. of the heat supplied also demonstrates that there are no excessive losses in continuous blow-down if the salinity is maintained at $\frac{3}{32}$ds.

Before leaving the subject of blow-down it may be stated that the only practical way of obtaining large capacities with a relatively small shell in any evaporator is by the use of continuous blow-down. The blow-down period for $\frac{1}{32}$d feed and $\frac{3}{32}$ds maximum salinity is obtained from the formula

* Wm. L. DeBaufre, Salt Water Evaporators, A. S. N. E. JOURNAL, November, 1915.

Item No	ITEM	UNIT	SOURCE							
1	Run Number		Test Data	1	2	3a	3b	4a	4b	4c
2	Date, 1917		Test Data	6-13	7-7	7-12	7-12	7-11	7-11	7-11
3	Time Of Start Of Run		Test Data	2:00	10:20	11:20	2:30	10:00	11:40	1:30
4	Time Of End Of Run		Test Data	3:30	11:30	1:20	3:30	11:30	12:40	2:30
5	Duration Of Run	Minutes	From 3 & 4	90	70	120	60	90	60	60
6	Number Of Rows Of Tubes		Test Data	6	6	6	6	6	6	6
7	Heating Surface Outside Of Tubes	Sq. Ft.	See Appendix I	26.67	26.67	26.67	26.67	26.67	26.67	26.67
8	Baffle Installed		Test Data	Yes	Yes	Yes	Yes	Yes	Yes	Yes
9	Water Level In Glass Above Bottom Of Bottom Tube	Inches	Test Data	14.0	14.0	12.0	12.0	10.0	10.0	10.0
10	Room Temperature	Deg. Fahr.	Test Data	88.0	84.0	81.0	81.0	80.0	80.0	80.0
11	Barometer	Inches Hg.	Test Data	30.22	30.16	29.70	29.68	29.74	29.79	29.71
12	Feed Temperature	Deg. Fahr.	Test Data	86.4	82.1	78.2	78.9	78.1	77.9	78.7
13	Feed Salinity	32nds	Test Data	0.25	0.23	0.23	0.23	0.23	0.23	0.23
14	Shell Salinity, Minimum	32nds	Test Data	3.06	2.76	2.91	3.06	2.86	2.96	3.06
15	Shell Salinity, Maximum	32nds	Test Data	3.86	2.86	3.06	3.16	3.06	3.06	3.06
16	Shell Salinity, Average	32nds	Test Data	3.27	2.86	3.06	3.06	3.03	3.09	3.06
17	Vapor Pressure in Shell, Gage	Lb. Per Sq. In.	Test Data	0.38	0.11	0.31	0.36	0.37	0.27	0.41
18	Vapor Temperature By Thermometer	Deg. Fahr.	Test Data	211.9	211.7	213.2	213.8	213.8	212.6	212.9
19	Vapor Temperature Corresponding To Press.	Deg. Fahr.	See Appendix I	213.7	212.7	212.7	212.8	212.9	212.6	213.0
20	Vapor Per Hour, Lb.	Lb. Per Hr.	See Appendix I	524.0	671.1	1424.0	1696.4	1738.7	1693.0	1771.0
21	Vapor Per Day, Gal.	Gal. Per Day	See Appendix I	1310.0	1891.0	3039.0	4108.0	5068.0	4181.0	5098.0
22	Chlorine Content Of Vapor	Gr. Per Gal.	Test Data	0.4	0.5	1.0	13.9	4.1	0.7	2.0
23	Heat Given To Vapor, Per Lb.	B.t.u. Per Lb.	See Appendix I	1096.6	1100.5	1104.4	1109.7	1104.6	1104.6	1104.0
24	Heat Given To Vapor, Per Hr.	B.t.u. Per Hr.	(23)X(20)	574618	738546	1572666	1806098	1842660	1604984	1956184
25	Steam Press. At Inlet, Gage	Lb. Per Sq. In.	Test Data	9.1	11.1	30.5	34.7	30.7	30.5	40.7
26	Steam Temp. At Inlet By Thermometer	Deg. Fahr.	Test Data	298.5	298.4	301.8	308.9	308.2	302.0	301.9
27	Steam Temp. At Inlet Corr. To Press.	Deg. Fahr.	See Appendix I	297.6	297.0	294.6	289.7	281.4	274.6	287.4
28	Superheat In Steam	Deg. Fahr.	(26)-(27)	1.9	30.4	18.2	16.2	26.8	27.4	13.9
29	Temp. At Coil Drain By Thermometer	Deg. Fahr.	Test Data	253.2	238.6	265.4	271.6	269.6	265.1	273.9
30	Steam Per Hour, Lb.	Lb. Per Hr.	See Appendix I	623.0	803.7	1667.8	1894.4	2044.0	1696.0	2068.0
31	Heat Abstracted From Steam, Per Lb.	B.t.u. Per Lb.	See Appendix I	992.6	968.0	951.9	946.9	948.3	951.9	938.8
32	Heat Abstracted From Steam, Per Hr.	B.t.u. Per Hr.	(31)X(30)	618184	777962	1587396	1793807	1940969	1613405	1940992
33	Heat Lost In Radiation, Blowdown And Unaccounted For	B.t.u. Per Hr.	(32)-(24)	43118	39496	14677	-12288	-2284	8421	5349
34	Per Cent Of Heat Lost And Unaccounted For	Per Cent	(33)÷(32)]100	2.9	5.1	0.9	-0.7	-0.1	0.5	0.3
35	Lb. Of Vapor Per Lb. Of Steam		(20)÷(30)	0.841	0.835	0.854	0.864	0.850	0.857	0.848
36	Temperature Difference Between Steam And Vapor Corr. To Press.	Deg. Fahr.	(27)-(19)	13.9	15.3	61.9	68.9	68.5	62.0	74.4
37	Lb. Of Vapor Per Hr. Per Sq. Ft. Of Surface Per Deg. Temp. Diff.		See Appendix I	0.022	0.029	0.060	0.078	0.068	0.079	0.059
38	Heat Transfer Coefficient, B.t.u. Per Sq. Ft. Of Surface Per Hr. Per Deg. Temp. Diff.		See Appendix I	901.8	948.2	992.6	968.8	1068.4	910.7	988.4

4b	4c	4d	5	6a	6b	6c	7a	7b	8	9a	9b	9c	9d	10a	10b	11a	11b
7-11	7-11	7-11	7-14	7-10	7-10	7-10	7-6	7-6	7-5	7-9	7-9	7-9	7-9	7-13	7-13	7-17	7-17
11:40	1:30	2:40	3:40	12:40	1:50	3:00	12:40	3:10	1:40	12:20	1:50	2:30	3:10	12:10	2:50	1:10	2:20
12:40	2:30	5:30	11:20	1:40	2:50	3:50	1:40	3:50	3:30	1:20	2:20	3:00	3:30	1:20	3:20	2:00	5:00
60	60	50	100	60	60	50	60	20	110	60	50	30	20	60	30	50	40
6	6	6	6	6	6	6	6	6	6	6	6	6	6	6	6	5	5
26.67	26.67	26.67	26.67	26.67	26.67	26.67	26.67	26.67	26.67	26.67	26.67	26.67	26.67	26.67	26.67	22.23	22.23
YES	YES	YES	YES	YES	YES	YES	YES	YES	YES	YES	YES	YES	YES	YES	YES	YES	YES
10.0	10.0	10.0	8.0	8.0	8.0	8.0	8.0	8.0	8.0	6.0	6.1	6.1	6.0	4.0	4.0	11.5	11.5
80.0	80.0	80.0	83.0	83.0	83.0	83.0	83.0	83.0	83.0	84.0	84.0	84.0	84.0	85.0	85.0	85.0	85.0
29.73	29.71	29.71	29.96	29.79	29.76	29.74	30.17	30.14	30.10	29.93	29.91	29.91	29.91	29.91	29.93	30.15	30.14
77.9	78.7	78.5	79.9	78.9	78.9	78.9	81.6	82.7	83.4	83.5	85.0	82.2	82.2	79.5	80.6	81.2	81.4
0.23	0.23	0.23	0.21	0.23	0.23	0.23	0.23	0.23	0.24	0.23	0.23	0.23	0.23	0.23	0.23	0.21	0.21
2.96	3.06	3.06	3.71	2.86	2.96	3.06	3.01	2.86	2.96	2.81	2.89	2.86	2.81	2.96	3.06	3.06	2.96
3.06	3.06	3.16	4.06	3.06	3.06	3.16	3.06	3.06	3.16	3.06	3.06	3.06	3.06	3.06	3.06	3.06	3.06
3.09	3.06	3.09	4.00	3.00	3.04	3.08	3.05	2.99	3.06	3.00	2.99	2.98	2.98	3.06	3.06	3.06	3.02
0.77	0.42	0.63	0.56	0.64	0.64	0.64	0.92	0.92	0.00	0.92	0.70	0.75	0.84	0.56	0.99	0.44	0.44
212.6	212.9	214.9	219.5	219.7	219.5	219.1	213.3	211.9	211.7	212.7	213.7	213.7	214.6	213.6	213.7	213.6	213.5
212.6	213.0	213.8	219.9	219.9	219.9	219.8	213.5	213.4	212.9	213.7	214.3	214.5	214.8	213.2	213.8	213.8	213.8
1859.0	1771.0	2140.9	1578.0	1895.0	1965.0	2048.0	1768.0	1859.2	681.9	1870.0	2228.0	2346.0	2466.2	1793.0	1874.0	1672.9	1748.9
5181.0	5038.0	6164.0	4543.0	5495.0	5634.0	5894.0	5080.0	5468.0	1814.0	5361.0	6413.0	6678.0	7098.0	4987.0	5368.0	4814.0	5038.0
0.7	3.0	18.7	0.7	3.1	8.0	7.0	1.6	2.8	0.4	1.9	4.8	3.9	7.2	1.6	4.8	3.9	6.8
1104.6	1104.0	1104.5	1102.9	1104.2	1104.1	1104.1	1101.5	1100.2	1103.0	1098.2	1098.2	1098.1	1098.2	1103.9	1097.3	1101.8	1101.6
2049.4	1989.184	2364.614	1741.479	2087.468	2169.557	2261.197	1948.200	2068.500	718.364	2058.604	2446.780	2630.011	2708.381	1913.018	2186.170	1849.181	1926.886
30.8	40.7	30.6	45.7	30.6	52.6	60.7	34.6	40.7	11.7	40.7	55.6	60.7	70.7	31.6	72.7	42.2	54.2
302.0	301.3	348.9	312.0	316.0	318.9	321.5	307.8	312.6	271.5	308.6	318.7	312.7	328.3	316.4	328.5	318.2	317.8
274.6	267.4	298.2	293.1	298.2	308.1	307.9	280.9	287.7	248.5	287.9	303.2	308.0	316.6	308.9	318.2	294.9	298.0
27.4	13.9	21.1	18.9	17.8	15.8	13.6	22.5	24.9	28.2	21.1	15.5	14.7	11.7	16.1	7.9	20.5	18.8
263.1	273.5	283.2	283.2	283.5	288.6	295.1	264.9	271.2	236.7	272.4	283.2	290.2	298.9	287.8	304.9	281.1	284.2
1996.0	2069.0	2838.1	1863.5	2250.0	2351.0	2444.0	2072.0	2211.2	793.8	2178.5	2638.0	2784.0	2967.3	2076.0	2398.0	1962.1	2108.9
954.5	938.8	957.7	994.7	955.0	991.0	986.0	958.9	948.4	868.0	943.5	934.5	928.8	922.5	991.2	944.4	988.9	988.2
15400	19605.7	15719.76	14181.5	21097.50	21701.61	21269.44	18716.461	21097.102	7129.78	20899.228	24830.78	25481.250	27371.934	19991.71	23193.492	18411.098	19712.348
8.421	5.940	15.382	9.94	11.291	6.04	19.67	28.284	7.601	11.294	4.124	12.228	11.231	28.985	21.192	27.266	-2.169	4.625
0.9	0.5	0.6	0.02	0.5	0.09	0.1	1.4	0.4	1.8	0.2	0.5	0.4	1.1	1.1	1.2	-0.1	1.8
0.857	0.849	0.849	0.847	0.842	0.848	0.838	0.854	0.859	0.854	0.858	0.847	0.848	0.834	0.858	0.852	0.858	0.858
63.0	74.4	84.4	73.8	84.3	88.2	94.1	66.8	74.3	51.0	73.8	88.9	98.8	101.8	87.1	104.4	81.1	88.2
0.878	0.895	0.891	0.742	0.848	0.896	0.816	0.928	0.926	0.788	0.880	0.880	0.894	0.908	0.786	0.748	0.898	0.913
910.7	988.4	1059.8	818.2	980.7	912.0	901.0	1093.5	1084.8	956.9	1044.4	1091.0	1014.0	987.6	913.1	882.4	1087.4	1017.2

PLATE I.

	4c	4d	5	6a	6b	6c	7a	7b	8	9a	9b	9c	9d	10a	10b	11a	
	7-11	7-11	7-14	7-10	7-10	7-10	7-6	7-6	7-5	7-9	7-9	7-9	7-9	7-13	7-13	7-17	
	1:30	2:40	9:40	12:40	1:50	5:00	12:40	9:10	1:40	12:20	1:50	2:30	3:10	12:20	2:50	1:10	
	2:30	5:30	11:20	1:40	2:50	3:50	1:40	9:30	3:30	1:20	2:20	3:00	3:30	1:20	3:20	2:00	
	60	50	100	60	60	50	60	20	110	60	30	30	20	60	30	30	
	6	6	6	6	6	6	6	6	6	6	6	6	6	6	6	5	
67	26.67	26.67	26.67	26.67	26.67	26.67	26.67	26.67	26.67	26.67	26.67	26.67	26.67	26.67	26.67	22.23	
	YES	YES	YES	YES	YES	YES	YES	YES	YES	YES	YES	YES	YES	YES	YES	YES	
0	10.0	10.0	8.0	8.0	8.0	8.0	8.0	8.0	8.0	6.0	6.1	6.1	6.0	4.0	4.0	11.5	
0	80.0	80.0	83.0	83.0	83.0	83.0	83.0	83.0	83.0	84.0	84.0	84.0	84.0	85.0	85.0	85.0	
79	29.71	29.71	29.96	29.79	29.76	29.74	30.17	30.14	30.10	29.99	29.92	29.92	29.92	29.92	29.99	30.13	
9	78.7	78.5	79.9	78.9	78.9	78.9	81.6	82.7	73.4	83.8	85.0	85.2	85.2	79.5	80.6	81.2	
23	0.23	0.23	0.21	0.23	0.23	0.23	0.23	0.23	0.24	0.23	0.23	0.23	0.23	0.23	0.23	0.21	
96	3.06	3.06	3.71	2.86	2.96	3.06	3.01	2.86	2.96	2.81	2.86	2.86	2.81	2.96	3.06	3.06	
06	3.06	3.16	4.06	3.06	3.06	3.16	3.06	3.06	3.16	3.06	3.06	3.06	3.06	3.06	3.06	3.06	
09	3.06	3.09	4.00	3.00	3.04	3.08	3.08	2.99	3.06	3.00	2.99	2.98	2.98	3.06	3.06	3.06	
27	0.41	0.43	0.36	0.64	0.64	0.64	0.32	0.32	0.00	0.32	0.70	0.73	0.84	0.36	0.39	0.44	
6	212.9	214.5	213.5	213.7	213.5	213.1	213.3	212.9	212.7	212.7	213.7	213.7	214.6	213.6	213.7	213.6	
6	213.0	213.8	213.3	213.9	213.9	213.8	213.5	213.4	212.9	213.7	214.5	214.5	214.8	213.2	213.8	213.8	
0	1771.0	2140.9	1578.0	1898.0	1968.0	2048.0	1768.0	1858.2	684.3	1870.0	2228.0	2364.0	2466.2	1732.0	1874.0	1672.9	
0	5098.0	6464.0	4543.0	5495.0	5694.0	5894.0	5090.0	5468.0	1874.0	5381.0	6413.0	6978.0	7098.0	4987.0	5366.0	4817.0	
7	3.0	18.7	0.7	3.1	8.0	7.0	1.6	2.8	0.4	1.9	4.8	8.8	7.1	1.6	4.8	3.5	
6	1104.0	1104.5	1102.9	1104.2	1104.1	1104.1	1101.3	1100.2	1103.0	1098.2	1098.2	1098.1	1098.2	1103.5	1101.9	1101.8	
84	1958184	2364624	1741479	2092459	2169557	2216197	1948200	2089500	718384	2089804	2446780	2830072	2708581	1911018	2186170	1849301	
5	40.7	50.6	43.7	50.6	55.6	60.7	54.6	40.7	11.7	40.7	55.6	60.7	70.7	51.6	72.7	47.2	
0	301.5	319.5	312.0	316.0	318.9	321.5	307.8	312.6	271.5	308.6	318.7	322.7	328.5	316.4	328.5	315.2	
6	287.4	298.2	295.1	298.2	303.1	307.9	280.3	287.7	243.3	287.9	303.2	308.0	316.6	300.2	318.2	294.9	
4	13.9	21.1	18.9	17.8	15.8	13.6	27.5	24.9	28.2	21.1	15.5	14.7	11.7	16.1	7.9	20.3	
1	272.5	283.2	283.2	283.5	288.6	293.1	264.9	271.2	258.7	272.4	285.2	290.2	298.9	287.8	304.9	281.1	
0	2089.0	2398.1	1863.5	2250.0	2331.0	2444.0	2072.0	2211.2	793.8	2173.5	2632.0	2754.0	2967.5	2076.0	2290.0	1962.1	
5	838.5	957.7	934.7	955.0	991.0	996.0	953.9	948.4	968.0	945.0	934.5	923.5	922.5	991.2	944.4	928.5	
05	1960527	2379976	1418615	2109750	2170161	2269144	1976481	2091102	729678	2099625	2488078	2541253	2797934	1999171	2309492	1841030	
421	5345	15382	934	11291	604	1947	28284	7601	11294	4134	12288	11431	28995	21892	27166	-2169	
9	0.5	0.6	0.02	0.5	0.09	0.1	1.4	0.4	1.8	0.2	0.5	0.4	1.1	1.1	1.2	-0.1	
857	0.848	0.849	0.847	0.842	0.843	0.832	0.854	0.859	0.854	0.858	0.847	0.849	0.851	0.855	0.850	0.858	
10	74.4	84.4	73.8	84.5	88.2	94.1	66.8	74.5	91.0	73.8	88.9	93.8	101.8	87.1	104.4	84.1	
679	0.885	0.891	0.742	0.848	0.896	0.816	0.955	0.956	0.788	0.880	0.940	0.914	0.908	0.966	0.968	0.956	
7	989.4	1059.5	848.2	930.7	912.0	901.0	1093.5	1054.8	868.9	1044.4	1091.0	1044.6	997.6	923.1	883.4	882.4	

PLATE I.

10b	11a	11b	12a	12b	13a	13b	14a	14b	15	16a	16b	17a	17b	18a	18b
7-15	7-17	7-17	7-18	7-18	7-20	7-20	7-23	7-23	7-25	8-1	8-1	8-3	8-3	8-6	8-6
1:50	1:10	2:20	12:00	2:00	12:50	2:50	11:48	1:08	1:08	12:15	2:25	10:35	1:25	10:35	1:09
2:20	2:00	3:00	1:00	2:50	1:50	3:50	12:48	2:08	2:08	1:15	2:09	11:48	2:25	11:35	2:09
40	30	40	60	30	60	40	60	60	60	60	60	40	30	60	60	60
5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5
22.67	22.23	22.23	22.23	22.23	22.23	22.23	22.23	22.23	22.23	22.23	22.23	22.23	22.23	22.23	22.23	22.23
225	YE5	YE5	YE5	YE5	YE5	YE5	YE5	YE5	YE5	M0	M0	M0	M0	M0	M0	M0
12.0	11.5	11.5	10.0	10.0	8.0	8.0	6.0	6.0	6.0	10.0	10.0	8.0	8.0	6.0	6.0	
85.0	85.0	85.0	85.0	85.0	86.0	86.0	85.0	86.0	89.0	86.0	88.0	85.0	89.0	85.0	85.0	
30.99	30.15	30.14	30.18	30.17	30.14	30.20	30.01	30.01	30.13	30.01	29.97	30.09	30.04	30.16	30.17	
81.6.	81.2	81.4	80.1	80.2	81.9	82.0	80.7	80.7	81.9	85.9	84.0	84.0	83.9	81.9	81.0	
0.23	0.21	0.21	0.21	0.21	0.23	0.23	0.22	0.22	0.21	0.23	0.23	0.23	0.23	0.23	0.23	
3.06	3.06	2.96	3.06	3.06	2.81	2.96	2.96	3.06	2.91	2.96	3.06	2.96	2.91	3.06	2.81	
3.06	3.06	3.06	3.16	3.06	3.06	3.16	3.16	3.06	3.26	3.06	3.16	3.06	3.91	3.16	3.91	
3.06	3.06	3.01	3.09	3.06	3.00	3.04	3.06	3.06	3.01	3.09	3.08	3.04	2.86	3.07	3.01	
0.53	0.44	0.46	0.44	0.54	0.54	0.64	0.44	0.64	14.20	0.49	0.95	0.17	0.97	0.15	0.91	
2.7	213.6	213.5	213.7	213.7	213.8	213.7	213.5	213.6	190.0	212.7	213.5	212.7	213.5	212.9	213.4	
2.8	213.8	213.8	213.9	214.2	214.9	214.6	213.6	213.5	168.8	213.8	213.2	212.8	213.4	212.8	213.5	
2.0	1672.9	1748.9	1706.0	1880.1	1799.0	2019.4	1772.0	2001.0	1844.0	920.0	1348.4	973.2	1996.0	1005.0	1494.0	
0.40	4817.0	5033.0	4910.0	5418.0	5160.0	5839.0	5100.0	5758.0	5694.0	2676.0	3881.0	2738.0	5734.0	2892.0	4238.0	
2.8	3.9	6.8	1.4	8.0	1.7	4.9	1.6	4.9	0.8	2.9	8.2	1.9	6.4	2.0	3.9	
2.5	1101.8	1101.6	1102.9	1102.8	1101.2	1101.5	1102.2	1102.5	1112.8	1099.1	1099.9	1098.6	1098.9	1100.7	1100.9	
1170	1869201	1926388	1881947	2084402	1744892	2134918	1893098	2106103	1689189	1021169	1481366	1066136	759844	1106204	1644748	
2.7	47.2	54.2	51.2	69.4	58.1	74.2	57.0	81.0	158.4	74.0	98.8	25.1	48.2	27.1	48.9	
2.5	918.2	917.8	918.9	919.0	921.8	928.5	917.9	919.1	968.0	886.0	906.7	887.8	909.6	886.9	911.1	
2.2	294.9	299.0	299.0	310.9	306.7	319.5	304.6	314.7	367.7	265.9	286.9	267.2	297.0	270.1	296.0	
2.9	20.9	18.8	16.9	12.9	15.1	9.0	12.9	4.4	0.5	21.7	19.2	20.6	6.6	16.8	15.1	
2.9	281.1	284.2	284.0	286.6	293.8	306.9	291.2	311.9	360.5	258.0	278.6	259.4	284.2	263.1	284.6	
2.0	1962.1	2108.9	2021.0	2271.7	2167.0	2479.4	2144.0	2480.0	2593.0	1089.0	1606.4	1142.4	1941.0	1174.0	1787.0	
2.4	928.5	928.2	921.8	923.1	923.7	919.1	926.8	926.7	863.4	854.2	940.4	921.5	928.8	946.5	921.4	
4.92	1841038	1972349	1883189	2097006	2008592	2269940	1987058	2148646	2236796	1048666	1510658	1088186	1801801	1171191	1666198	
2.26	-2166	48628	9642	12604	21540	28962	20961	41395	75819	16593	28092	18978	48987	4987	21484	
2.2	-0.1	1.9	0.2	0.6	1.6	1.5	1.7	1.9	3.4	2.6	1.9	1.7	2.8	0.5	1.9	
0.830	0.888	0.898	0.844	0.891	0.827	0.869	0.896	0.807	0.730	0.846	0.858	0.884	0.823	0.894	0.896	
2.4	84.1	85.2	85.1	96.9	97.4	104.9	91.0	111.4	117.9	51.9	71.3	54.4	88.6	57.9	81.5	
1.748	0.886	0.913	0.901	0.888	0.873	0.870	0.876	0.808	0.742	0.891	0.851	0.905	0.898	0.788	0.869	
2.4	1021.4	1017.2	994.6	978.7	964.2	928.4	963.5	880.9	818.4	891.9	993.4	884.1	945.7	868.4	896.8	

$$\frac{V}{32} + \frac{WP}{32} = \frac{3V}{32}, \text{ or, } P = \frac{2V}{W},$$

in which

V = Volume of shell to the operating water level, expressed in pounds of water;

P = Blow-down period, hours;

W = Capacity of evaporator, pounds of vapor per hour.

In order to obtain the rated capacity of 1,300 pounds per hour with the evaporator tested, using intermittent blow-down and a maximum salinity of $\frac{3}{32}$ds, the blow-down would have to occur at 30-minute intervals. The practical impossibility of operating with this blow-down period is evident.

The proper water level and steam pressure at which to operate any evaporator to obtain maximum capacity may be determined by making runs at constant water levels and slowly increasing steam pressure and at the same time measuring the chlorine content of the distillate according to the methods used in this test. Particular attention is called to the necessity of providing means for venting the air from the steam manifold and drain pot, and allowing a small amount of steam to blow from these vents continuously to insure the removal of air.

APPENDIX I.

Giving the source of items in table of results which are not explained in table.

Item.

7. Heating surface outside of tubes, square feet.
   From: Area of one row of tubes multiplied by number of rows of tubes (Item 6).
   Where area of one row of tubes 1¼ inches O. D. and total length 15.09 feet equals 4.444 square feet.

19. Vapor temperature corresponding to pressure, degrees F.
   From steam tables corresponding to absolute vapor pressure. (Item 17 + (item 11 × 0.49117)).

20. Vapor per hour, pounds.
   From: Total vapor per run divided by duration of run in minutes (item 5) multiplied by 60.

21. Vapor per day, gallons.
Vapor per hour (item 20) divided by 8.34 multiplied by 24, where 8.34 = pounds of water per gallon.

23. Heat given to vapor, B.t.u. per pound.
Total heat in one pound vapor (from steam tables corresponding to absolute vapor pressure) minus the heat in the feed (heat of the liquid corresponding to feed temperature, item 12).

27. Steam temperature at inlet corresponding to pressure, degrees F. From steam tables corresponding to absolute steam pressure. Item 25 + (item 11 × 0.49117).

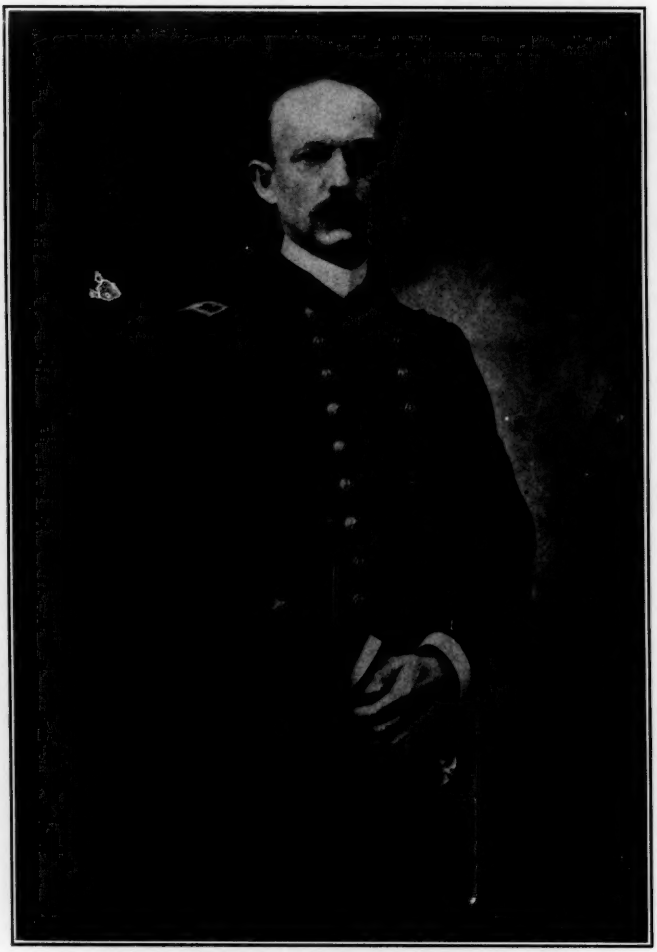
30. Steam per hour, pounds.
Total steam during run divided by duration of run in minutes (item 5) multiplied by 60.

31. Heat abstracted from the steam, B.t.u. per pound.
Total heat of inlet steam (from pressure and superheat on Ellenwood steam charts) minus heat in coil drain (heat of liquid at drain temperature, item 29).

37. Pounds of vapor per hour per square foot of surface per degree temperature difference.
Pounds of vapor per hour (item 20) divided by square feet of surface (item 7) × temperature difference (item 36).

38. Heat transfer coefficient, B.t.u. per square foot of surface per hour per degree temperature difference (item 23 × item 37).

NOTE.—All thermometer readings were corrected for calibration error and exposed steam. All gage readings were corrected for calibration error and water in syphon.



CAPTAIN EMIL THEISS, U. S. NAVY

CAPTAIN EMIL THEISS, U. S. NAVY.

The Service as a whole, and its engineering branch particularly, suffered a real loss in the untimely death on September 24, 1917, of Captain Emil Theiss. For himself and his friends there was a special touch of sadness in that his physical condition had prevented his promotion to the grade of Rear Admiral, for which he had been recommended by the Board of Selection.

Captain Theiss was born in Hattingen, Germany, October 2, 1860, but accompanied his parents to this country at an early age. He was admitted to the Naval Academy as a Cadet Engineer, by competitive examination, October 1, 1878, and began to show at once the ability for which he was distinguished throughout his entire naval career. He graduated in 1882 at the head of his class with one of the highest averages which had been attained up to that time. His first duty was on the U. S. S. *Tennessee*, from 1882 to 1884. He was promoted to Assistant Engineer while on this duty.

He graduated just in time to take part in the first amalgamation of the Line and the Engineer Corps, which occurred in 1882, so that for several years he performed the combined duties very much as is the case today. It was subsequently decided by the courts that the cadet engineers who had left Annapolis prior to the Act of August 5, 1882, were not subject to that Act, and this restored Captain Theiss to the Engineer Corps, which was the natural field for his talents. The rest of his career in the Navy was as an engineer, and he was recognized throughout the Service as one of the ablest men in the Corps.

His abilities as a designing engineer became evident at an early period, with the result that he was on duty more than once in the Bureau of Steam Engineering as one of the principal assistants to such able designers as Towne, Canaga and Bailey. It is not going too far to say that the success of the

machinery of our naval vessels owes much to his skill and care in looking after the details of the design.

He carried out successfully his tours of duty at sea, serving on the *Tennessee*, *Ranger*, *Monterey*, *Albatross*, *Kearsarge*, *Alabama*, *Galveston* and *Iowa*, and when his rank precluded further sea duty he gave much attention to inspection work. His last active duty, in fact, was as the Engineer Member of the Naval Inspection Board, where his extended experience, ripe judgment and great ability enabled him to render most valuable service. He was selected to go abroad and study foreign methods of engineering administration.

His record was so fine that, when the Board of Admirals appointed to select officers for promotion made their report, he was recommended for one of the permanent appointments as Rear Admiral. He had been so active in the performance of duty that his friends did not suspect (nor probably did he) that he could not pass the physical examination for promotion. This proved, however, to be the fact, and he was granted sick leave with the idea that he might recuperate. It was decided finally that an operation was desirable, but the results were unsuccessful and he died soon after the operation.

Captain Theiss was a man of agreeable personality and counted a large circle of friends, both inside and outside the Service. The writer, as an associate for several years on duty in the Bureau of Steam Engineering and a friend from Naval Academy days, remembers well his spirit of helpfulness and good cheer, which made him always an agreeable colleague. He was frank and outspoken in his opinions, but modest, as able men usually are, so that he was ready to discuss questions that seemed debatable and to consider the views of others.

The writer well remembers a little incident which occurred at the beginning of the war with Spain, when he and Theiss were on duty in the Bureau under that Grand Old Chief, Admiral Melville. Theiss came into the Admiral's office, while the writer happened to be present, and said to him, "I would like to be detached, sir, and ordered to sea duty." "What is is the matter," said the Admiral, "are you not satisfied with

your duty and your treatment?" "Oh, yes, sir. I like the work very well, indeed, but my training has been to be a fighting man in time of war, and this seems to be the only chance that I'll ever have to see any real fighting." To this the Admiral's answer was, "My son, I appreciate your position, which is exactly what it ought to be. Every officer worth his salt ought to want to go to the front. However, some of us must stay here and do the work which is required; I can't get away and neither can you. I could easily fill the places of fifty men on board ship, but I would not know where to turn to fill your place here." It is a pleasure to place this pleasant little episode on record to show that Theiss was at heart a "Fighting Engineer."

It is particularly fitting that this sketch of Captain Theiss should appear in this JOURNAL, and the notice would be sadly incomplete without mention of the great part which he played in the success of the JOURNAL. He was one of the very active workers in the preparation of its pages from the very start, and he was the third Editor, the first being Admiral Griffin, and the second, the writer. Captain Theiss was unusually well equipped for this work from his natural ability, his accomplishments as an engineer and his skill as a linguist. The numbers of the JOURNAL which he edited show better than any eulogy how well his work was done, but I do not believe that any of the other editors will feel envious when it is said that he was one of the best editors the JOURNAL ever had. He was one of the founders and a charter member of this Society.

In reflecting upon Theiss' career the writer's thoughts are carried back to the days when he and the rest of us were all enthusiastic youngsters working for Melville, whom we all worshiped as a man and loved like a father. He had the genius to develop the greatest ability that each man possessed, and we all counted it a privilege to work for him without regard for hours, carrying out his maxim, "we are all working to make our Navy the best in the world."

WALTER M. MCFARLAND.

THEISS OF THE *MONTEREY*.

When the battle bugles play,
And the battle-flags are flying,
Mid mad music of the fray,
'Tis not hard for soldiers—dying.

But when none is there to tell
Who would crave to dare or die?
Who would face the fires of hell—
Asking not—nor caring—why?

It was but the other day
One there was who knew not fear,
On the war-ship *Monterey*,
Theiss, the gallant engineer.

His the best and bravest daring;
When the war-ship's boiler burst,
Not for gain or glory caring,
He was in the death-cloud first.

With the bravest volunteer
Who has gone his hero's way,
Rank him—Theiss, the engineer
Of the warship *Monterey*.

With his comrades of the crew,
Strong of heart and firm of lip,
While the death blast round them blew,
There they stayed to save the ship.

Of such stuff are heroes made;
With a will they worked away,
Life was duty, death their trade;
But they saved the *Monterey*.

Nail our banner to the mast;
Of such spirits brave and bold,
He was neither first nor last;—
We've the metal and the mold!

From "An Inn for Journeying Thoughts;" WILLIAM J. ROE.

NOTES.

METALLOGRAPHY.

The following very interesting article was originally read before the IRON AND STEEL INSTITUTE and later published in the "Ironmonger."

THE "GROWTH" OF CAST IRONS AFTER REPEATED HEATINGS.

BY PROFESSOR H. F. RUGAN, MEMBER A. S. M. E.; TULANE UNIVERSITY OF LOUISIANA, U. S. A.; RESEARCH FELLOW IN METALLURGY, VICTORIA UNIVERSITY, MANCHESTER; AND PROFESSOR H. C. H. CARPENTER, M. A., PH. D., PROFESSOR OF METALLURGY, VICTORIA UNIVERSITY, MANCHESTER.

INTRODUCTION.

The fact that certain types of cast iron "grow" after repeated heatings has long been familiar to engineers. Cast-iron annealing ovens 8 feet in length, 3 feet in diameter, and 1½ inches in thickness, which are kept red hot for prolonged periods, sometimes "grow" to 9 feet in length in the course of use. Cast-iron furnace grates, ranges, fittings, &c., which are subject to alternate heating and cooling, are frequently distorted and sometimes broken from the same cause. The Baltimore fire is of particular interest in this connection, because what had previously been considered acceptable fire-proof construction was proved to be unsatisfactory owing to damage done to the masonry by the "growth" of the cast-iron members of the ironwork construction. Floor levels were destroyed, and walls which had in other respects withstood the action of the fire were found to be unsuited for further use owing to their having been forced out of the perpendicular from this cause.

The same fire also furnished an illustration of the fact that under the same treatment mild steel does not grow. In spite of the entire inflammable contents of a sixteen-story building being burned, the steel framework remained "plumb and intact." Other instances of a similar kind have been recorded.

Further, it is well known that wrought-iron contracts permanently after repeated heatings. Mr. Humphrey Wingfield has suggested that these might be related phenomena, and that it would be interesting to discover an alloy intermediate between cast iron on the one hand, and mild steel and wrought iron on the other, which neither expanded nor contracted after repeated heatings. This problem, which was first studied by Mr. A. E. Outerbridge, Jr., of Philadelphia, has been taken up by the authors, and the results of their investigations are recorded in the following paper.

EXPERIMENTS TO DETERMINE THE CONDITIONS UNDER WHICH THE MAXIMUM GROWTH OF CAST IRON TAKES PLACE.

A grey cast iron was selected for this purpose. Bars of known length were heated in an electrically-warmed tube furnace at certain temperatures and for certain periods, the temperature being measured by a plati-

num platinum-iridium thermo-junction and direct-reading portable pyrometer. Three bars were heated at a given temperature. One was removed after one hour, another after two hours, and the third after three or, in some cases, seventeen hours. When cold they were remeasured. The following table summarizes the results obtained:—

TABLE I.

Heat No.	Temperature, degrees C.	Bar No. 1.		Bar No. 2.		Bar No. 3.	
		Time.	Growth in inches.	Time.	Growth in inches.	Time.	Growth in inches.
1	600	1 hour.	nil.	2 hours.	nil.	3 hours.	nil.
2	650	1 hour.	nil.	2 hours.	0.0020	3 hours.	0.0050
3	675	1 hour.	0.0156	2 hours.	0.0250	3 hours.	0.0300
4	700	1 hour.	0.0234	2 hours.	0.0342	3 hours.	0.0312
5	730	1 hour.	0.0312	2 hours.	0.0346	3 hours.	0.0352
6	760	1 hour.	0.0313	2 hours.	0.0350	17 hours.	0.0352
7	800	1 hour.	0.0313	2 hours.	0.0350	17 hours.	0.0352
8	860	1 hour.	0.0313	2 hours.	0.0350	17 hours.	0.0352
9	900	1 hour.	0.0315	2 hours.	0.0352	17 hours.	0.0355

Under the conditions of the experiments no growth was observed at 600 degrees C. (though, if the heating be sufficiently prolonged, growth takes place at much lower temperature; *e. g.*, cast-iron valves subjected to superheated steam at 366 degrees C. increase in size permanently. The length of time required for this, however, is such as to suggest that the phenomena are different). A slight expansion was evident after two hours' heating at 650 degrees C., which was notably increased after another hour's heating at that temperature. At 675 degrees C. the rate of growth, for a given period, increased about eight times . At 700 degrees and 730 degrees there were also substantial increases; but maximum growth was virtually reached in three hours at the latter temperature. The same expansion was obtained at 730 degrees as at 860 degrees C. The figures also show that three hours is a sufficient period of heating, no further growth being observed after seventeen hours. A very slight, but only just perceptible, increase in growth was caused at 900 degrees C. As this temperature is doubtless reached, and perhaps to some extent exceeded, in some of the practical uses of cast-iron annealing ovens, the authors selected it for their later experiments, and in order to ensure maximum growth for a given heat the period of heating chosen was four hours.

METHODS OF MEASUREMENT.

The first measurements were made with commercial cast-iron bars, on which two points at a known distance from one another had been marked, the disatnce between the points being measured as the heats progressed. This method, however, was soon found to be unsatisfactory. After a number of heats at 900 degrees C. a scale formed on the bars which rendered the marks indistinct and consequently the measurements uncertain.

It was decided to machine the bars accurately to a given length and diameter and to measure their expansion by micrometer calipers, each recording differences of 0.0001 of an inch. The retort in which the test bars were heated was of grey cast iron, and itself furnishes an illustration of the distortion of this material after repeated heatings, this

particular retort having been heated upwards of fifty times to about 900 degrees C. Originally the bottom of the retort was flat. It was screwed on to a wrought-iron plate with a layer of asbestos in between to assist in the distribution of the heat inside the retort. In the center was a cast-iron tube for the insertion of the thermo-junction used for temperature measurements.

The lid was luted into the retort with clay to diminish air access. The packed retort was placed in a clay muffle, heated by gas externally in a large Fletcher furnace, the specimens being thus completely protected from the direct action of the flame. The results, nevertheless, indicate that the products of combustion of the flame do enter the retorts and cause a scale to form on the specimens. It is, however, after a considerable number of heatings that the accuracy of measurements is interfered with.

As a rule one heat per day was carried through. After the temperature in the retort had reached 850 degrees to 900 degrees C. it was maintained there for four hours. The bars always cooled at approximately the same rate, and they were not measured till next day, the greatest care being taken that they were then at the normal laboratory temperature.

SERIES OF EXPERIMENTS WITH COMMERCIAL CAST IRONS.

Three irons were selected.

The first was a mixture used by a firm for the manufacture of annealing ovens. The second was a mixture as far removed from the first in composition as they used. Bars of these were cast in flasks of green sand and dry sand, respectively. Before pouring, the dry-sand flasks were heated by bars of red-hot iron. One flask of each kind was poured from the same ladle, the only difference being that the bars cooled at a different rate. These bars are designated PH (hot), PC (cold), FH, and FC. The third sample (NIP) was a bar cut from a new annealing oven obtained from another source.

These five samples were heated ninety-nine times for four hours at about 850 degrees to 950 degrees C., the following summary showing the main features of the growths observed.

TABLE II.

Alloy.	Percentage.		Percentage increase of volume.				Percentage increase of weight after 99 heats.
	Carbon.	Silicon.	After 10 heats.	After 21 heats.	After 52 heats.	After 99 heats.	
PC	3.48	1.13	19.90	27.80	32.40	37.50	8.60
PH	3.48	1.13	19.80	27.46	31.50	36.60*	8.50
FC	3.41	0.95	8.71	18.25	26.55	35.21	7.86
FH	3.41	0.95	13.05	22.45	28.10	35.70	7.82
NIP	3 14	0.96	19.30	23.15	27.70	36.80	7.90

*Percentage growth after ninety-four heats.

So far as carbon and silicon are concerned, the compositions of the three alloys were not greatly different. Alloys PC, PH, and NIP grew rapidly and reached 50 per cent. of their ultimate growth after ten heats. After twenty-one heats their rate of growth became very slow. Alloys

FC and FH grew more slowly at the outset, but ultimately nearly caught up the others. The rate of growth of PC and PH was very similar throughout, whereas FH, a more "open" iron than FC, grew more rapidly at the outset. In both series, however, the ultimate growth was nearly the same, irrespective of the initial rate of cooling of the alloy, varying from 35.21 to 37.50 per cent.

Although the ends of the bars grew rather more rapidly than the sides, and numerous small cracks formed on the surface as the heats progressed, their ultimate shape was cylindrical.

An increase in the weight of the bars also took place, the rate being more uniform than in the case of the volume increase. The percentage increase after ninety-nine heats varied from 7.82 to 8.60.

The results of this series of experiments are in general similar to those obtained by Mr. Outerbridge, with one important difference, viz: that instead of the weight of the enlarged bar being precisely the same as before treatment, the authors found a progressive increase in weight as growth proceeded.

Long before these experiments were concluded it had become clear that the first step to be taken in the elucidation of the cause or causes at work during the growth of cast iron by repeated heatings was to correlate "growth" with the composition of the material. It was, of course, necessary to begin with simpler materials than the complicated cast irons of commerce, and the first series of alloys was designed so as to contain, as nearly as industrial processes permit, only iron and carbon.

IRON–CARBON SERIES.

The compositions of the iron-carbon series are given in Table II.

TABLE II.

Alloy.	Percentage.				
	Carbon.	Silicon.	Manganese.	Sulphur.	Phosphorus.
A	4.03	0.243	0.15	0.018	0.010
B	3.70	0.233	0.16	0.018	0.011
C	3.08	0.172	0.13	0.019	0.010
D	2.59	0.146	0.12	0.016	0.012
E	1.96	0.215	0.20	0.013	0.015
F	1.58	0.168	0.17	0.015	0.014
G	0.99	0.233	0.19	0.015	0.012
H	0.55	0.182	0.20	0.013	0.014
I	0.15	0.186	0.15	0.040	0.017

The carbon descends broadly in half per cents. from 4 to the lowest carbon alloy that could be cast (0.15). The remaining constituents are present in sufficiently constant and, in the case of sulphur and phosphorus, small amounts. The rather higher figure for sulphur in alloy I was caused by the greater time needed for melting. This series proved quite satisfactory for the purpose for which it was intended.

Each alloy was cast both in sand and chill molds, the diameter of the bars being 1 inch. About 25 pounds of each were available. In no case was any graphite present; the alloys were exclusively white irons.

Machined bars of these alloys were subjected to as many as ninety heats in some cases. The chill castings were too hard for turning. Approximately a 6-inch length was ground in the lathe, the bar being

held in a chuck while an emery-wheel was fed across the end surface. This gave a suitable form for length measurements. Near the center of this bar a space of about 1 inch was ground to a true surface and served for diametric measurements. The remainder of the surface was left as cast.

These samples retained their shape throughout the heat treatment. In the case of alloy A the thickness of the scale after ninety heats was about 0.0015 inch, and tended to increase as the carbon of the series diminished.

The sand castings were less hard. They were turned to about 0.75 inch diameter in the form of 6-inch lengths. Alloys E, F, G, and H contained a number of blowholes.

The maximum temperature of the various heats was 960 degrees C., the minimum 860 degrees C.

Complete details of the volume and weight measurements made from time to time were kept, a summary of the ultimate changes of volume and weight after the last heat being shown in Tables III and IV. A plus quantity indicates an increase, and a minus quantity a decrease of volume or weight.

TABLE III.—CHILL CAST ALLOYS.

Alloy.	Number of heats.	Percentage change of volume.	Percentage change of weight.
A	90	+6.880	—1.21
B	90	+0.238	—1.28
C	87	—0.840	—1.29
D	87	—0.822	—0.97
E	87	+1.140*	—0.75
F	48	+0.290*	—0.43
G	48	+1.350*	—0.12
H	48	+0.113*	—0.02
I	48	+0.800*	+0.03

* These figures are high, on account of the scale which formed to a greater extent on the lower carbon members of the series. The same holds for the sand castings.

TABLE IV.—SAND CAST ALLOYS.

Alloy.	Number of heats.	Percentage change of volume.	Percentage change of weight.
B	78	—0.350	—1.72
C	78	—1.675	—1.19
D	78	—0.915	—1.19
E	78	—0.026	—1.47
F	39	—0.405	—0.64
G	39	—0.470	—0.52
H	34	—0.507	—0.17
I	39	—0.060	+0.02

The sand-cast alloys, which did not include A, not only did not grow under this treatment, but for the most part showed a progressive shrinkage. In the case of D, E and I, there was a momentary expansion as a result of the first or second heats, but it was followed by a contraction. In some cases the shrinkage reached a maximum value and then dimin-

ished. In others (*e. g.* C) it progressed almost uninterruptedly through seventy-eight heats. In all cases the final result was a permanent shrinkage, which, in six out of the eight alloys, did not reach more than half 1 per cent.

A progressive diminution in weight took place throughout, except in the case of alloy I, where it was succeeded by an increase after the twenty-ninth heat.

Alloys B to I of the chill-cast series behaved similarly to those of the sand-cast series, even although there was a permanent shrinkage in only two out of the eight alloys. Alloys B to E all showed a shrinkage at the sixty-seventh heat. The apparent slight permanent expansions recorded from E to I in the summary are probably caused by the scale, which naturally forms more plentifully on the lower-carbon members, and tends to loosen itself from the bar, thus giving too high a reading.

The only alloy which showed a permanent growth of any magnitude was A, and even this showed a shrinkage after thirty-four heats. A permanent expansion was recorded at the thirty-eighth heat, and in the next fifty-two heats the bar grew slowly but steadily until, at the ninetieth heat, a growth of 6.88 per cent. was obtained, the explanation of this behavior being found at a later stage.

Broadly speaking, pure iron-carbon alloys, whether chill or sand cast, do not expand permanently after repeated heatings, but, on the contrary, shrink. The amount of shrinkage is so small that, from a practical standpoint, it is almost if not quite negligible. The results obtained at this stage gave the authors the solution of their practical problem, viz: an alloy whose volume shows no tendency to change appreciably after numerous heatings to about 900 degrees C. Such an alloy would be a white iron with about 3 per cent. of carbon and only small quantities of other constituents, silicon being most important. This alloy would shrink slightly on repeated heatings, but probably not more than about 1 per cent. after one hundred heats. This type of shrinkage is usually ascribed to the gradual removal of contraction stresses set up in the originally cast metal. The loss of weight observed, which tends to diminish with decreasing carbon content, is caused by the gradual oxidation of carbon by the furnace gases which penetrate into the muffle and into the bars themselves. The real is no doubt greater than the apparent loss, because there is probably a simultaneous absorption of oxygen, which is evidently the explanation of the increase in weight of alloy I.

When it was found that alloys A to I, whether cast in chill or sand molds, were white irons, four more alloys were made in which the carbon ranged from 4.6 to 3.0 per cent., and the silicon varied between 0.5 and 0.79 per cent. They also contained a considerable amount of manganese. They were cast in sand, the base of this series being a grey iron. The following is the complete analysis made:

HIGH-CARBON MEDIUM SILICON SERIES.

...	Percentage.				
	Carbon.	Silicon.	Manganese.	Sulphur.	Phosphorus.
J	4.60	0.79	1.26	0.018	0.016
K	3.90	0.69	1.09	0.019	0.021
L	3.60	0.65	1.02	0.018	0.017
M	3.02	0.50	0.86	0.018	0.017

The fracture of J showed a grey iron, except for a mottled area near the center. K and L were almost entirely white irons, and contained only small amounts of graphite. M was a white iron. Test bars were ground to size in the same way as those of the A to I chill series. They were heated sixty-eight times. These alloys grew steadily from the outset. Ultimate growth was reached in from fifty to sixty heats; it was greatest in J and least in L. The following table enables a comparison to be made:

TABLE V.

Alloy.	Percentage growth after			Percentage ultimate growth.	Percentage change of weight.
	10 heats.	19 heats.	29 heats.		
J	3.48	8.02	11.26	12.65	+ 0.21
K	2.35	6.10	7.80	10.35	— 0.74
L	2.24	5.15	6.68	8.70	— 1.05

After reaching a maximum growth the bars showed a tendency to contract slightly. The bar with the least silicon shows the least growth, and that with the highest silicon the greatest growth. Alloy M behaved differently from the above trio, and in a manner which recalled that of A under similar conditions. It contracted during the first ten heats, but after the eleventh heat it showed a growth, and this increased until a maximum value, viz: 6.2 per cent., was obtained after fifty-eight heats. The test bar of M had been cut from a part near the gate end of the casting. In order to see whether the same behavior would be shown by a bar (MM) cut from another part of the casting, the top of one of the risers was selected, *i. e.*, a part as far distant as possible from the other. In this case the bar contracted up to the ninth heat. It was not measured after the tenth heat, but after the eleventh heat it was found to be expanding. The rate, however, was slower than in the case of M, and it was only after the fourteenth heat that the bar showed a growth as compared with the original size. The rate of growth remained slower, and the maximum growth, obtained after forty-seven heats, was only 2.54 per cent.

Apart from the difference in the amount of the ultimate growth, the fact of a contraction up to the ninth or tenth heat, followed by an expansion, was thus confirmed.

To account for this behavior of alloys A and M, a hypothesis naturally suggested itself in view of the two main facts hitherto established by the research. These facts were:

1. All the alloys examined which did not contain graphite or free carbon, in some form, did not grow, but actually contracted to a greater or less extent (alloys B to I).

2. All the alloys examined which contained graphite grew (alloys PH, NIP, J, K and L).

The natural hypothesis to account for the behavior of A and M was to assume that these alloys, originally white irons, contracted as long as they remained white, but that after a time they deposited free carbon, probably as temper carbon, and that this, either directly or indirectly, caused their growth. This hypothesis was tested by heating short pieces of bars A and M along with the main bars. Samples for chemical analysis were turned from these pieces, beginning with a heat at which contraction was still taking place, and following the heats through the point at which contraction ceased and expansion began. Each sample

was tested for temper carbon, and if a positive result was obtained, a quantitative estimation was made by the usual method.

In the case of alloy M, no temper carbon was present up to and including the ninth heat. Beyond this it was readily detected, and it increased with the progression of the heats, as shown by the following figures:

No. of heat.	Volume change.	Percentage temper carbon.
1–9	Shrinkage.....	nil.
10	Growth.	0.049
12	More growth.....	0.203
19	Still more growth.....	0.689

With alloy A no temper carbon was detected up to the eighteenth heat. It was found in minute amount after the nineteenth heat, and increased in the twenty-first and twenty-second heats, the absolute quantities, however, being much smaller than in the case of M. A comparison with alloy A showed that up to the nineteenth heat this latter contracted, and that after this the contraction diminished, *i. e.*, a slight expansion set in, followed by a contraction between the twenty-fourth and twenty-sixth heats, and again succeeded by an expansion, which, however, only attained a positive value at the thirty-eighth heat. Obviously the rate of formation of temper carbon was much slower than in M, and, while it does not furnish so good a test, it bears out the results obtained with that alloy.

The change from shrinkage to growth under repeated heatings in the case of alloys A and M was thus shown to coincide with the appearance of free carbon, probably temper carbon, which was in this way proved to be an indispensable factor in the growth of cast iron under these conditions. This fact is of fundamental importance.

It being evident that the influence of silicon must be thoroughly investigated, the next series of alloys was designed so as to contain a constant quantity of carbon (about 4 per cent.), and ascending percentages of silicon from 1 to 6. Manganese, sulphur and phosphorus were to be kept as low as possible. The reason for this choice was that the earlier members of the series would contain some "combined" carbon, and that the later would contain all their carbon in the "graphitic" condition; by repeated heating, the former would tend to deposit their combined as "temper" carbon, and if the "growth of cast iron" were due to an expansion consequent on this chemical change, the earlier members should grow and the later ones should not. Such a series would also test whether growth was proportional to the percentage of silicon.

The following calculation gives an idea of the volume change that may be expected to occur when cementite decomposes into ferrite and temper carbon. The uncertainty attaching to the calculation arises from the absence of accurate data on the specific gravity of "temper carbon." Assuming that the specific gravity is the same as that of graphite, the volume change can be deduced as follows:

Specific gravity of Fe_3C at 16 degrees C. = 7.07.
Specific gravity of Fe at 16 degrees C. = 7.86.
Specific gravity of graphite at 17 degrees C. = 2.10.

$$Fe_3C \longrightarrow 3Fe + C.$$

Weight (grams) 180 —> 168 + 12.
Volume (cubic centimeters) 25.16 —> 21.37 + 5.71,
i.e., 25.46 cubic centimeters become 27.08 cubic centimeters.
Volume increase = 6.36 per cent. . . . . (A)

If, however, the specific gravity of temper carbon be less than that of graphite, and approximates more to the pulverulent form of carbon, viz: charcoal, a higher value is obtained. Taking 1.45, the lowest figure obtained for charcoal, the maximum value for the expansion consequent on this change is obtained thus:

$$Fe_3C - 3Fe + C.$$

Weight (grams) $180 - 168 + 12.$

Volume (cubic centimeters) $25.46 - 21.37 + 8.28,$

i.e., 25.46 cubic centimeters become 29.65.

Volume increase = 16.46 per cent. . . . . (B)

These calculations do not give more than the order of magnitude of the volume increase, and they refer, of course, to 100 per cent. Fe_3C, which not even a white iron contains. They are, however, important, because they may account for the growths of A and M observed.

A contained originally 4.03 per cent. of combined carbon, which corresponds to about 60.45 per cent. Fe_3C. If a complete conversion of this to iron and temper carbon be assumed after ninety heats, then,

According to calculation A there would be a growth of 3.84 per cent.

According to calculation B there would be a growth of 9.95 per cent.

The found value, 6.88 per cent., is exactly the mean of the two (6.89).

Again, M contained 3.02 per cent. combined carbon, corresponding to 45.3 per cent. Fe_3C. Assuming a complete decomposition, then,

According to calculation A there would be a growth of 2.88 per cent.

According to calculation B there would be a growth of 7.45 per cent.

The maximum growth found for M was 6.20, and for MM 2.54 per cent. These values differ somewhat widely, but that for M falls within, and that for MM well below, the two calculated values.

Accordingly it does not appear that the influence of this change on the growth of white irons, which become grey on heating, can be disregarded, though when more accurate data are available, the above calculations may need considerable revision.

IRON–CARBON–SILICON SERIES.

The analysis of the alloys designed to test more specifically the part played by silicon is as follows:

TABLE VI.

Alloy.	Percentage.						
	Total carbon.	Combined carbon.	Graphite.	Silicon.	Manganese.	Sulphur.	Phosphorus.
N	3.98	0.64	3.34	1.07	0.25	0.010	0.013
O	3.98	0.68	3.30	1.79	0.23	0.010	0.013
P	3.79	0.30	3.49	2.96	0.25	0.010	0.012
Q	3.76	nil.	3.76	4.20	0.27	0.010	0.012
R	3.79	nil.	3.79	4.83	0.30	0.010	0.012
S	3.38	nil.	3.38	6.14	0.30	0.010	0.013

The total carbon in the series is approximately constant; alloys N and O contain about the same amount of combined carbon; alloy P contains about half the quantity, and the remaining alloys contain none at all. The silicon in O and R is 0.2 per cent. lower, in Q 0.2 per cent. higher

than was desired. The remaining constituents are satisfactorily low and constant.

This series has furnished the most information as to the cause or causes at work in the permanent expansion of cast irons after repeated heatings.

FIRST SERIES OF EXPERIMENTS WITH ALLOYS N TO S.

Test pieces, 6 inches by about 0.88 of an inch, taken haphazard, were machined from the castings. When the growth of these alloys was investigated it quickly became evident that the position from which a test piece had been cut had a considerable influence on its rate of expansion, specimens taken from the gate end of the casting growing more rapidly than those taken from the top of a riser. The behavior of the six specimens is shown in the following summary:

Alloy *N* (test piece cut from a riser at some distance from the gate).—This was an iron with a fine, close grain. Under heat treatment it grew steadily, and reached its maximum growth in about sixteen heats. It did not crack, and retained its cylindrical form to the end.

Alloy *O* (position from which the test piece was cut is uncertain).—This iron was very similar to N, except that the grain was coarser. Slight cracks developed over the entire surface after the twelfth heat, enlarging somewhat after the fourteenth heat, but thereafter remaining comparatively constant. Maximum growth was reached in about sixteen heats.

Alloy *P* (test piece cut from the bottom of the gate).—This appeared to be a very "open" iron, but retained its cylindrical shape almost perfectly throughout the heat treatment, and did not crack. It grew very rapidly at the outset—25.2 per cent. in five heats—after which the rate diminished considerably. Maximum growth was reached in about sixteen heats.

Alloy *Q* (test piece cut from a riser).—This was an open-grained iron. After the third heat it was noticed that the ends were larger than the central part of the sample; but at a later stage the center "caught up" this increased growth. This fact was noticed with other specimens. After the sixth heat the bar was bent; after the eighth it was badly bent. Cracks developed at the twelfth heat, and these continued to enlarge. After the fifteenth heat accurate measurements were impossible to carry out, but the heats were continued, because the bar increased in weight. After fifty-two heats an approximate determination of the growth was obtained by measuring the circumference of the bar and making an allowance for the cracks. The result showed that very little growth had taken place after the fifteenth heat. This bar grew much more slowly than P.

Alloy *R* (test piece cut from a riser).—This alloy was very similar in appearance to Q, and behaved very similarly under heat treatment. Cracks appeared at the twelfth heat, and accurate measurements had to be abandoned after the thirteenth heat. An approximate measurement was made after the fifty-second heat.

Alloy *S* (test piece cut from the bottom of the gate).—After the first heat a circular crack developed at one end, the outer 1/8-inch circumference cracking away from the core. After the second heat enormous cracks developed, the largest being 5/64-inch wide. After the sixth heat this had increased to about 1/4 inch, and the sample broke in two pieces at the center while being handled. Thirty-two heats were carried through, and small particles constantly disengaged themselves from the cracked surfaces, thus rendering the measurements of weight too low. An approximate volume determination (probably within 5 per cent. of the exact value) was made by the method indicated under Q.

Complete measurements of the volume changes show the rate of growth, which is rapid at first, but diminishes after about the seventh heat, and stops at the sixteenth heat.

N, O and P kept their cylindrical form in spite of a growth of 32.85 per cent. in the last case, and that Q, R, and S increasingly distorted under the heat treatment given. The following table summarizes the results:

Alloy.	Percentage silicon.	Percentage growth on heating.
N	1.07	15.40
O	1.79	23.46
P	2.96	32.85
Q	4.20	43.9
R	4.83	59.5
S	6.14	63.0

These tests show that silicon is a most important constituent of cast iron from the standpoint of growth under repeated heatings, that growth being proportional to the percentage of silicon, although the value for R seems relatively higher than the others. The growths are far in excess of those that could be caused by the conversion of combined into free carbon. Alloy N contained originally 0.64 per cent. combined carbon, corresponding to 9.6 per cent. Fe_3C. The complete decomposition of this in accordance with

$$A \text{ would cause growth of } 0.61 \text{ per cent.},$$
$$B \text{ would cause growth of } 1.58 \text{ per cent.},$$

both inadequate to account for the 15.4 per cent. growth found. The same obviously holds even more strongly for alloys O and P.

The fact that the two alloys P and S, which had been cut from the gate, grew more rapidly than the remaining four, which had not, pointed to the agency of the gas contained in the original casting, and perhaps also the gases in contact with the test pieces in the muffle. The gate of a casting is, as a rule, more "open" and contains more gas than the risers; and the next step consisted in testing bars comparable with one another as regards gas content, and taken from the top of a riser. This series is indicated with double letters, except in the case of O, which is marked with a triple letter.

DOUBLE-LETTERED SERIES, NN TO SS. VOLUME CHANGES.

These samples grew under heat treatment to approximately the same ultimate growth as the corresponding members of the previous series, but the rate of growth was in all cases slower. Cracks did not develop in QQ and RR until after the eighteenth heat, as compared with twelve heats in the single-letter series. But the most marked difference was shown by SS, which did not develop cracks until after the eighteenth heat, whereas S cracked after the first heat. Further, although SS cracked extensively before ultimate growth was reached, the cracks never became so large, nor was the specimen as much deformed. These results can be most simply explained by supposing that the castings contain their dissolved gas unevenly distributed, the gate containing the maximum and the top of a riser the minimum, and that the disruptive effects noticed in S were due to an excessive amount of dissolved gas.

The end of a sample nearest the gate always grew more rapidly during the earlier heats than the other end, this being afterwards equalized and even reversed, as shown in the following table:

Heat No.	Alloy OOO.		Alloy QQ.		Alloy RR.	
	Far.	Near.	Far.	Near.	Far.	Near.
	Per cent.	Per cent.	Per cent.	Per cent.	Per cent.	Per cent.
3	1.92	2.44	1.53	1.65	1.54	1.94
10	6.30	6.08	7.40	6.83	7.37	6.73
15	7.28	6.55	8.20	8.20	9.15	7.98

NOTE.—The words "Far" and "Near" refer to positions with reference to the gate.

The more rapid growth of the end nearest the gate is possibly due to a larger amount of dissolved gas. The fact that the other end ultimately caught up and sometimes passed it, led the authors to attribute this to the action of gases in contact with the specimens in the muffle.

The following table enables a comparison of the ultimate growths of the two series to be made with each other and also with the percentages of silicon they contain:

Percentage silicon.	Percentage.	Growth on heating.
1.07	N 15.40	NN 16.80
1.79	O 23.46	OOO 24.60
2.96	P 32.85	PP 34.40
4.20	Q 43.9	QQ 41.6
4.83	R 59.5	RR 55.3
6.14	S 63	SS 61

Although the double-lettered series grew more slowly than the previous one, the final results are in good agreement, particularly in the case of the first three members, where greater accuracy of measurement was possible. From the two series of tests the authors draw the following conclusions:

1. In a series of cast irons of approximately constant carbon the ultimate growth after repeated heatings is, broadly speaking, proportional to the silicon present. This relationship holds up to 6 per cent. of silicon.

2. The growth is far in excess of that which can be caused by the conversion of combined to free carbon, and is, in fact, much the largest in high silicon alloys which contain no carbon combined with iron. (Alloys Q, R and S.)

3. The behavior of the alloys on repeated heating, and particularly of Q, R and S, points to the view that two agencies are at work in their growth, viz:

(*a*) The internal pressure of gas which they contain in their original cast condition, and probably chiefly physical in its character.

(*b*) The effect of external gases in contact with the alloys during their heat treatment, and probably partly physical and partly chemical in its nature.

As regards 3 (*b*) considerable light has been thrown on the matter by a consideration of the changes of weight which accompany the changes of volume discussed above.

THE INCREASE IN WEIGHT ACCOMPANYING THE GROWTH OF THE ALLOYS.

SERIES N TO S.

Alloys N, O and P increase in weight rapidly up to about the tenth heat, after which they decrease somewhat and reach a nearly constant weight in about twenty-six heats. Alloys Q and R increase in weight rapidly up to the fifteenth heat, after which the rate is diminished up to the thirtieth heat, and then increasing at an accelerated rate up to the fifty-second heat. Just as in the case of the increase in volume, Q and R increase in weight more slowly than P up to the eleventh and ninth heat respectively. The increase in weight of S is throughout the largest of the series.

SERIES NN TO SS.

This series increased in volume more slowly than the other, and the increase in weight was also less rapid, notably with QQ, RR and SS, especially the last named, where the rate of increase was almost uniform.

The following table enables a comparison to be made between the final increases in weight obtained in the two series:

Percentage silicon.	Percentage increase in weight	
	After twenty-six heats.	
1.07	N 0.65	NN 0.94
1.79	O 1.89	OOO 2.24
2.96	P 3.62	PP 4.03
	After fifty-two heats.	
4.20	Q 9.59	QQ 9.33
4.83	R 10.25	RR 10.30
614	S 13.20	SS 11.55

The increase in weight (like the increase in volume) follows the same order as the increase in silicon. There are, however, complications in this case which probably do not affect the volume changes and make it more difficult to interpret the "increase in weight" curves.

During the heating and cooling of the alloys two causes are at work, one of which tends to increase, the other to decrease, their weight. They absorb gases from the interior of the muffle, principally carbon dioxide, carbon monoxide, and nitrogen, the two first named tending to oxidize the silicon and carbon in the alloys. The former process, of course, results in an increase, the latter in a decrease of weight. That these two processes actually take place was shown in the following way:

The Oxidation of Silicon.—The fact that the increase in weight follows the same order as the increase in silicon at once establishes a presumption in favor of this, especially as this element is the only variable in the series apart from iron, which, of course, correspondingly decreases. Analyses of silicon were made, beginning with the original casting, and continuing until the specimen of maximum weight was reached. There does not seem to be any thoroughly reliable chemical method of distinguishing between combined and dissolved silicon and silica in cast irons. It was found, however, on analyzing any one series of the six alloys, that a sample of the original casting dissolved completely in hydrochloric acid, and that the final precipitate of silica after ignition was quite flocculent and white. Proceeding

through the series, there came a point at which hydrochloric acid failed to dissolve the entire sample, a microcrystalline gray residue being left. The final precipitate of silica was less flocculent and had a greyish tinge. Proceeding further, the insoluble residue increased until in the fully grown sample it reached a maximum, and the final precipitate was neither white nor flocculent but a grey microcrystalline mass of very small bulk. These results seem to indicate that in the original sample the silicon was unoxidized and doubtless present as iron silicide in solid solution in the remainder of the iron. This would dissolve completely in hydrochloric acid and give ultimately a white amorphous flocculent precipitate. The grey microcrystalline residue, insoluble in hydrochloric acid, is probably silica deposited from solid solution by the action of the oxides of carbon at the temperature of the experiments. Evidence of these microcrystalline particles was also obtained in the microscopic examination of the alloys.

The Oxidation of Carbon.—This was shown by the following analyses made on average samples:

Alloy.	Percentage carbon in the original alloy.	Percentage carbon after twenty-five heats.	Percentage loss.
N	3.98	2.45	38.4
O	3.98	2.40	39.7
P	3.79	2.60	31.4
Q	3.76	2.77	26.3
R	3.79	3.25	14.3
S	3.38	3.44	nil.

The loss is greatest in N and O, which is only to be expected, as these contain the greatest amount of combined carbon—a more easily oxidized form than graphite. It diminishes with rising silicon until in S it is nil.

The increase in weight of the alloys observed is the net result of these two processes working in opposite directions. The oxidation of silicon takes place much more rapidly than that of carbon, particularly in the case of N, O and P; and it is the latter process which causes the drop in the curves after twelve heats. The results leave no doubt that, with the exception of S, the apparent increases in weight are less than the real. An approximation to the latter can be reached in the following way:

Alloy.	Mean apparent percentage increase in weight	Percentage loss of carbon.	Approximate percentage gain in weight.
N	0.85	1.53	2.38
O	2.07	1.58	3.65
P	3.85	1.19	5.04
Q	5.99	0.99	6.98
R	6.66	0.54	7.20
S	8.97	nil	8.97

The mean apparent increase after twenty-five heats is calculated, and to this is added the loss in carbon, after the same number of heats, the results obtained being those shown in the previous table.

If the increase in weight from complete oxidation of the iron silicide to silica and iron be calculated, the following results are obtained, enabling a comparison to be made with the above approximate increase:

Alloy.	Percentage silicon.	Approximate percentage gain in weight after twenty-five heats.	Calculated gain on assumption $FeSi \rightarrow SiO_2 + Fe$.
N	1.07	2.38	1.22
O	1.79	3.65	2.04
P	2.96	5.04	3.37
Q	4.20	6.98	4.78
R	4 83	7.20	5.51
S	6.14	8.97	7.00

These figures show that something besides the silicon in the alloys is oxidized, even after only twenty-five heats, which tends to cause an increase in weight, and it is clear that this must be iron itself, nothing else being present in sufficient quantity to account for the difference. The researches of Mr. Guillet indicate that the silicon in alloys of this type is present as iron silicide, FeSi, in solid solution. The most likely supposition, then, is that this compound is oxidized to a mixture of iron oxide and silica under the conditions of the experiments.

Even this, however, is insufficient to account for the whole of the increase in weight, as the following table shows:

Alloy.	Percentage silicon.	Approximate percentage gain in weight.	Calculated gain on the assumption. $FeSi \rightarrow FeO + SiO_2$.
		After twenty-five heats.	
N	1.07	2.38	1.83
O	1.79	3.65	3.07
P	2.96	5.04	5.07
		After fifty-two heats.	
Q	4.20	10.45	7.20
R	4.83	10.81	8.28
S	6.13	12.38	10.42

It appears, therefore, that in addition there is some oxidation of the iron, uncombined with silicon, except possibly in the case of P. This deduction agrees with the fact that the fully grown alloys have very largely lost their metallic nature, and cut more like a piece of chalk than a piece of metal, their micro-structure being also revolutionized during growth.

The study of the changes of weight accompanying the growth of these alloys has thus shown:

A partial oxidation of carbon, which diminishes with increase of silicon and becomes nil at 6 per cent.; a probably complete oxidation of the silicon, originally present as iron silicide, to a mixture of iron oxide and silica; and a partial increase as iron uncombined with silicon. It is quite clear that the increase in volume and in weight are closely related to one another, and that the growth of these alloys under heat treatment is bound

up with a change in the chemical condition of the silicon and to some extent of the iron.

MICROSCOPIC STUDY OF THE ALLOYS.

A very large number of specimens were investigated in this part of the research, alloy P being selected as the middle member of the series with about 3 per cent. of silicon. A number of sections, about 0.25 of an inch thick, were cut from the original bar and heated in the muffle. From time to time as the heat progressed one of these was withdrawn, and in this way a series of sections was obtained at various stages of growth. The designation of these and the number of heats to which they correspond are contained in the following table:

Designation of section		The number of heats to which it was exposed.
Alloy P	0	Original metal
	1	1
	2	3
	3	6
	4	10
	5	15
	6	21

No. 0 corresponds to the original casting, No. 6 to the fully grown alloy, and the intervening numbers to the intermediate stages of growth. No. 0 contained a few small holes. These increased somewhat in 1 and 2, the increase being mainly from the outside. In the remainder the holes were more uniformly distributed, but increased in number and size from No. 3 to No. 6.

The sections were then etched with 1 per cent. nitric acid in absolute alcohol. The surfaces of Nos. 0 and 1 dissolved quietly. No. 2 gave a slight evolution of gas, which became more marked in No. 3 and was very large in No. 4, but diminished in No. 5 and was still less in No. 6. These results show that as the heats progressed the alloy absorbed gas, which increased up to about the tenth heat, and after that diminished. The appearance of the etched surfaces at this stage gave a most interesting idea of the way in which gases penetrated into the alloy.

No. 0 showed the etched surface of the original alloy. In No. 1 there was a thin, dark ring at the edge, that is believed to indicate the depth to which gases penetrated after the first heat. This altered layer etched more readily than the rest of the alloys, hence the dark appearance. No. 2 (after three heats) showed a considerably deeper penetration of gas, however, some distance from the center, as indicated by concentric rings of various degrees of darkness. In No. 3 the gases appeared to have reached the center, and the dark rings were very marked. These diminished in Nos. 4 and 5 and were entirely absent from No. 6, owing, no doubt, to the fact that after twenty-one heats the chemical and physical effects of the penetrating gases were completed, and the material, being as homogeneous as No. 0, etched uniformly.

The sections were then examined at a magnification of 150 diameters, and typical structures photographed.

In the original casting the greater part of the carbon is present as graphite (seen as black lines), the remainder of the field (mottled) showing the solid solution of iron silicides in the residue of the iron, together with a small quantity of combined carbon.

Next a boundary between a gas-penetrated area (dark) and a still unaffected part of the interior (light), in which the graphite plates and the solid solution were clearly shown.

The next showed a completely gas-penetrated area in the early stages (after three heats).

An examination was made of the structure of a part of the interior to which the gas had not yet penetrated after three heats. It was interesting. because it threw light on changes in the interior of the alloy before the gases penetrating from the outside had time to make their influence felt. Lines distributed through the white areas, although they doubtless corresponded to original positions of graphite, no longer represented as much of it, being for the most part slits from which the graphite plates had been removed. The agency in this removal appeared to be the gas originally dissolved in the metal, and probably liberated by heating at the surface of the solid solution in contact with the graphite. The latter is much the weakest mechanical constituent, and it appears to be forced into holes originally present in the casting and enlarged by the breaking down of the boundaries between two holes, when these also consist of graphite.

A typical structure after maximum growth had been reached bore a distinct resemblance to the original metal, with the important difference that graphite had vanished from its original position in the form of plates, and the sharp, black lines were replaced by blurred markings. These represented areas oxidized by the incoming hot gases which passed along the slits caused by the removal of graphite under the influence of the pressure of the originally dissolved gases. Although large holes were numerous in the fully grown alloy, a field free from holes was selected to illustrate the final structure, as it enabled the resemblance and the contrast to the original alloy to be most readily seen.

Before proceeding to study the influence of gases on the growth of the alloys, one question still remained to be answered. Although graphite, or temper carbon, had been found in all the alloys that grew on heating, yet in the nearly constant graphite series, N to S, the growth had been found to be roughly proportional to the silicon present. To settle the question whether an iron-silicon series containing no graphite would grow on repeated heating, three alloys with ascending silicon and as little carbon as possible were cast under similar conditions to previous alloys. Their analysis gave the following results:

IRON-SILICON SERIES.

Mark.	Percentage.				
	Silicon.	Carbon.	Manganese.	Sulphur.	Phosphorus.
T	0.65	0.17	0.17	0.045	0.017
U	1.10	0.18	0.19	0.049	0.022
V	2.71	0.19	0.20	0.051	0.033

Microscopic examination showed in all three cases a solid solution (of iron silicide in iron), with no traces of graphite or any other structural constituent.

Machined bars were heated fifteen times under the same conditions as previous alloys. A summary of the changes of volume and weight are given in the table below:

Alloy.	Percentage silicon.	Percentage change of volume after fifteen heats.	Percentage change of weight after fifteen heats.
T	0.65	— 0.025	— 0.04
U	1.10	0.000	— 0.03
V	2.71	+ 0.394	— 0.02

Alloy T expanded slightly, after the first heat, remained constant after the next three, and then contracted slightly. There was also a slight progressive diminution in weight. Alloy U expanded very slightly after the first heat, returned to its original volume after the second heat, and then remained steady. The diminution in weight was very similar to that of T. Alloy V, after five heats, had still its original volume, and then began to grow very slightly, but even after fifteen heats the growth was only 0.394 per cent. It decreased in weight the least of the three alloys.

Thus the only alloy of the three which showed any tendency to grow was V, with 2.71 per cent. of silicon. The expansion was, however, very slight, and, compared with that of P (2.96 per cent. silicon and 3.79 per cent. carbon) after the same number of heats, was almost negligible (0.394 as compared with the mean figure of P and PP, 31.35 per cent.).

Alloys K, N and P correspond closely to alloys T, U and V in silicon content. They also contain about 3.9 per cent. of carbon, mostly in the form of graphite, as compared with a mean of 0.18 per cent. of carbon, none of which is present as graphite, in the other series. A comparison can thus be made between the changes of volume of the two series under similar tests after fifteen heats by means of the following table:

Alloy.	Carbon.	Silicon.	Percentage change of volume.	Alloy.	Carbon.	Silicon.	Percentage change of volume.
T	0.17	0.65	— 0.025	K	3.90	0.69	+ 5.40
U	0.18	1.10	0.000	N	3.98	1.07	+ 15.20
V	0.19	2.71	+ 0.394	P	3.79	2.96	+ 31.35

This comparison emphasizes that free carbon, either as graphite or temper carbon, is an essential factor in the growth of cast irons under heat treatment. The previous series of alloys, however (N to S), also clearly showed that in a constant graphite series the growth is roughly proportional to the silicon present. It is clear, therefore, that both graphite and silicon are involved in these changes of volume after repeated heatings.

INVESTIGATION OF THE INFLUENCE OF GASES ON THE GROWTH OF CAST IRONS.

The apparatus used consisted of a glazed Berlin porcelain tube, which would hold a vacuum for many hours at 900 degrees C., when heated by gas. The ends were closed with purified rubber stoppers. On the left-hand side of the tube a stopcock was inserted through the stopper and served to introduce air into the apparatus when required. The specimen to be tested was introduced through this end. An arrangement for water cooling the ends of the tube in contact with the stoppers was also used. The other end of the tube communicated with a mercury gage and a gas pipette of known volume, and thence to a Fleuss' pump. The same period

of heating, viz: four hours, was allowed as in the gas-heated muffle experiments. The specimens of alloy for test were machined to 6 inches by ½ inch.

EXPERIMENTS WITH S.

This alloy was chosen because it had shown the maximum growth of the N to S series, viz: about 62 per cent., and a 6-inch length was machined as close as possible to the gate of the casting, in order to secure a piece comparable with the original specimen of S, which cracked so quickly on heating in a muffle. This specimen is designated S½. Its exact volume was 19.88 cubic centimeters.

This specimen was heated in the evacuated apparatus, the temperature being raised gradually. Gas was given off readily (the exact temperature was not measured). After two heats the bar was measured and weighed and its appearance noted.

ALLOY S, HEATED IN VACUO AT 900 DEGREES C.

Length, inches.	Diameter, inch.	Volume, cubic ins.	Change, per cent.	Weight, grams.	Change, per cent.	Heat No.
6.0250	0.5227	1.2928	.....	144.816	.....	.....
6.0167	0.5234	1.2943	+0.120	144.821	+0.003	2
6.0157	0.5234	1.2941	+0.101	144.820	+0.003	5
6.0145	0.5233	1.2924	−0.04	.....	.....	11
6.0145	0.5233	1.2924	−0.04	144.828	+0.008	16

The above table gives the exact data. A slight growth was found (0.12 per cent.), and a minute increase in weight (0.003 per cent.). Twelve and four-tenths cubic centimeters of gas had been evolved at this stage. The surface of the bar was slightly discolored (greenish-yellow to blue), this being more marked at the gate end. The authors interpret this as indicating that the gas evolved from the metal, from about 800 degrees C. upwards, reacted with it, causing the discoloration, and that the effect was more marked at the gate end because more gas was present. After the fifth heat the bar had begun to shrink, but was still a little larger than the original size. After the eleventh heat it had shrunk further, and was 0.04 per cent. smaller than the original. This value was not altered after sixteen heats. The evolution of gas continued over eight heats (thirty-two hours), and reached a total of 22.20 cubic centimeters. The bar, therefore, gave up 1.11 times its own volume of gas.

It was not possible in this experiment to determine the average composition of the gas evolved, since the apparatus had to be disconnected from time to time to enable the bar to be measured and weighed. The determination was, however, made on another sample cut somewhat farther from the gate, and it was noticed that much less gas was given off. Its composition was as follows:

Constituent.	Percentage by volume.
Carbon dioxide	nil
Carbon monoxide	nil
Oxygen	nil
Hydrogen	87.5
Nitrogen	12.5

Thus the only constituents were hydrogen and nitrogen, present in the proportion of 7 to 1 by volume. This result agrees with that obtained by Troost and Hautefeuille, who found that hydrogen was the principal gaseous constituent of cast irons.

During the sixty-four hours' heating the bar gained 12 milligrams in weight (0.008 per cent.). This is very small, and was probably caused by a very slight leak in the apparatus.

The experiment thus showed that alloy S, when heated to 900 degrees C. in vacuo, actually shrinks slightly in length and expands in width. This alloy grew 62 per cent. (mean value) when heated to constant volume in a muffle in contact with gases, thus proving that the growth is caused by gases.

A study of the volume changes enables a further important conclusion to be drawn, viz: that the gas contained in the alloy cannot have any appreciable effect in causing the growth of this alloy, otherwise there would be a permanent growth in vacuo. The experiment has shown that it is possible to extract this gas without causing more than a slight temporary expansion. It is therefore clear that the external gases in the muffle are responsible for the 62 per cent. growth of S.

TABLE LX.—ALLOY SS¼.

FIRST HEATED IN VACUO AT 900 DEGREES C. TO EXTRACT DISSOLVED GASES, AND AFTERWARDS IN THE MUFFLE FURNACE.

Length, inches.	Diameter, inch.	Volume, cubic ins.	Growth, per cent.	Weight, grams.	Increase, per cent.	Heat, No.
6.0180	0.5271	1.313	.....	146.879	.....	.....
6.0210	0.5278	1.318	0.380	146.936	0.44	2
6.0250	0.5288	1.320	0.765	147.041	0.11	6
6.0600	0.5328	1.350	2.82	147.542	0.45	12
6.0880	0.5365	1.373	4.08	.....	.....	21
6 0930	0.5371	1.380	5.12	148.020	0.78	22
6.1010	0.5374	1.388	5.73	148.112	0.84	23
6.1125	0.5390	1.395	6.25	148.266	0.94	24
6.1200	0.5390	1.400	6.64	148.361	1.01	25
6.1270	0.5398	1.405	7.02	148.462	1.08	26
6.1400	0.5410	1.420	8.19	148.608	1.18	27
6.1525	0.5423	1.425	8.59	148.784	1.31	28
6.1695	0.5442	1.435	9.29	149.000	1.45	29
6.1870	0.5455	1.440	9 70	149.180	1.58	30
6.2260	0.5491	1.480	12.95	149.660	1.90	32
6.3010	0.5563	1 535	17.95	150.490	2.46	35
6.4600	0.5703	1.679	28.00	151.433	3.11	37
6.5870	0.5828	1.760	33.75	152.363	3.73	39
6.6650	0.5892	1.815	38.30	153.244	4.34	41
6 8125	0.6053	1.960	49.40	154.532	5.21	43
6 9375	0.6150	2.060	57.00	156.440	6.55	45
7.0312	0.6194	2.118	61.30	160.490	9.30	46
7.1171	0.6270	2.140	63.00	162.333	10.50	48
7.1250	0.6288	2.212	67.70	165.245	12.50	53
7.1250	0.6288	2.212	67 70	164.890	12.26	58

Experiment has substantiated this deduction conclusively in the following way: A fresh bar of alloy S was machined to about 6 inches by 0.5 of an inch. It was heated in vacuo at 900 degrees C. until no more gas was

given off, and afterwards in the muffle under the same conditions as the other bars.

This bar (SS½) grew very slowly. After twenty-one heats the expansion was only 4.08 per cent., and after thirty only 9.7 per cent. Beyond this stage, however, it grew rapidly. After fifty-three heats the growth was 67.7 per cent., which is, in fact, slightly larger than the figures 63 and 61 obtained for S and SS, respectively. After five more heats there was no further growth. The growth of alloy S is therefore due entirely to the agency of gases penetrating from the muffle.

There was, however, a most important difference between the final condition of SS and S, on the one hand, and SS½, on the other. Whereas the two former cracked extensively during growth and became greatly deformed, the latter hardly cracked at all, and kept its cylindrical form almost perfectly. The cracking of S and SS, therefore, is due to pressure from within caused by the dissolved gases hydrogen and nitrogen, their coefficient of expansion at the ordinary temperature being about 400 times greater than that of grey cast iron. The rapid growth is due to the relatively large surface thus exposed to the penetration of the muffle gases.

At the conclusion of its growth SS½ was again heated in vacuo and the evolved gas analyzed. It had the following composition:

Constituent.	Percentage by volume.
Carbon dioxide	33.00
Carbon monoxide	14.54
Oxygen	nil
Hydrogen	42.50
Nitrogen	9.96

showing that nearly half the total quantity of penetrating gases consists of oxides of carbon, which are responsible for the oxidation of the iron silicide so closely bound up with the growth of the alloy.

The proof just given that the growth of this alloy is caused entirely by the penetration of gases from the muffle is important, because it explains the interesting fact that the corners not only of this alloy but also of Q and R grew more rapidly than the rest of the bar, the ends being cupped.

There can be no doubt that the more rapid growth of the end of the bar nearest the gate of the casting is connected with the fact that it contains more dissolved gas. But this does not of itself explain the more rapid growth, because the experiments in vacuo have shown that this gas can be removed on heating without causing any permanent growth. Although time has not permitted this question to be investigated, the authors think that the more rapid growth, which is usually accompanied by more or less cracking, may be caused by an explosive reaction between the penetrating oxides of carbon and the dissolved hydrogen in the alloy.

EXPERIMENTS WITH N.

This alloy was chosen because it presented the greatest contrast to S. It was a very close-grained iron, had the greatest mechanical strength of the series, and grew the least on heating. The results obtained, while apparently not so striking as those with S, are equally interesting and scientifically important.

A bar (N½) cut from the gate end of the casting was machined to the usual size, and heated in vacuo under similar conditions to S½. Gas was given off, but much less readily. After the first heat the bar was measured, and found to have increased 1.63 per cent. in volume. The growth proceeded steadily up to twenty-five heats, when a maximum was reached, no further increase taking place after an additional ten heats. The ultimate growth was 11.1 per cent.

After the first heat a loss in weight of 3 milligrams was found. Beyond this, however, a gain in weight took place of the same order of magnitude as with S (26 milligrams in thirty-five heats), and doubtless also due to a minute leak through the stoppers.

The total volume of gas evolved was 23.27 cubic centimeters, the volume of the bar being 19.87 cubic centimeters. The bar thus gave up 1.17 times its volume of gas. Its composition, determined on another sample obtained from a fresh bar, was as follows:

Constituent.	Percentage by volume.
Carbon dioxide	1.09
Carbon monoxide	nil
Oxygen	nil
Hydrogen	98.54
Nitrogen	0.37

The gas thus consisted almost entirely of hydrogen.

The experiment thus showed that this alloy behaves quite differently from S½ when heated in vacuo, and that it grows to a very considerable extent, both absolutely and relatively to the total growth when heated in a muffle.

In order to get an exact comparison another bar, NN½, was heated in vacuo until no further gas was evolved, then transferred to a muffle and heated to ultimate growth under the same conditions as SS½. It grew very similarly to N and NN, although its ultimate growth was rather less, viz: 13.50 per cent. The growths of N½ and NN½ were similar, although NN½ grew rather more rapidly, but the final difference between them was only 2.4 per cent., whereas between the corresponding S½ and SS½ it was 67.7 per cent.

The fully grown NN½ was finally heated in vacuo and the evolved gas analyzed:

Constituent.	Percentage by volume.
Carbon dioxide	17.61
Carbon monoxide	27.46
Oxygen	0.35
Hydrogen	40.14
Nitrogen	14.44

As in the fully grown SS½, the gases which penetrated into NN½ consist of oxides of carbon to the extent of about 45 per cent.

The behavior of alloy N is thus in complete contrast to that of S, as is shown in the following table:

Alloy.	Percentage growth when heated in vacuo.	Percentage growth, when heated in muffle furnace after previviously extracting as much gas as possible in vacuo.
N	+11.10	+13.50
S	−0.04	+67.70

The most striking difference is the behavior when heated to constant volume in vacuo. S shrinks slightly, but remains almost constant, whereas N grows 11.1 per cent. To explain this difference the authors suggest the

following hypothesis: S is a very open-grained alloy, and the gas it contains in its originally cast condition will be comparatively readily removed by heating in vacuo without causing a permanent growth. On the other hand, N is a close-grained alloy, a compact aggregate of iron silicide (dissolved in iron) and graphite with very few holes, and the gas it contains is removed in vacuo with much greater difficulty. In fact, the only explanation of the 11 per cent. growth in vacuo is that some of the gas remains permanently imprisoned in the alloy, its enormously greater coefficient of expansion causing an internal pressure sufficient to expand the metal permanently. In this way cavities are formed, and doubtless enlarged in places where the boundaries between two cavities consist of mechanically weak graphite.

(The expansion due to complete conversion of combined to free carbon in N cannot, at the most, account for more than 1.58 out of a total of 11.1 per cent.)

N thus appears to furnish an instance of an alloy in which the dissolved gases are capable of causing a large growth, at any rate when external gases are removed.

On the other hand, when N is caused to grow by heating in a muffle, the curves of increase of volume and increase of weight leave no doubt that they are closely related. The greater part of the increase is effected in the first ten heats, the growth being probably largely due to the penetrating gases. How much is due to the pressure of originally dissolved gases it is very difficult, if not impossible, to estimate, but since the enlargement of the ends of bars N, O and P in the early stages was very much less marked than in Q, R and S, the pressure of the dissolved gas probably is responsible for the more even growth in these cases in some way that is yet not understood.

SUMMARY.

Summarizing these results, it may be said that there are alloys, *e. g.*, S, whose growth in air is due entirely to oxidizing gases penetrating their interior, and others, *e. g.*, N, where this may not be the sole agent, but where originally dissolved gases contribute to some extent to the growth.

Iron-carbon Series of Alloys Containing No Graphite.—The carbon ranged between 4.03 and 0.15 per cent. Other constituents were low and constant. (Eight alloys, B to I.)

A. Sand-Cast Alloys.—White irons. The final result after thirty-nine to seventy-eight heats was a permanent shrinkage in all cases, which in six out of eight cases was not more than 0.5 per cent. A diminution in weight was found in seven out of eight cases, which followed nearly the same order as the carbon.

B. Chill Cast Alloys.—White irons. The results were similar to those obtained with the sand cast alloys, although the permanent shrinkage was not so fully marked.

The above results have gvien the solution of the practical problem, viz: the finding of an alloy whose volume remains constant even after repeated heatings at about 900 degrees C. Such an alloy would be a white iron with about 3 per cent. of carbon, and only small quantities of other constituents of which for this purpose silicon is the most important and should not exceed about 0.2 to 0.3 per cent.

High-Carbon Medium Silicon Series (J, K and L), Containing Small Amounts of Graphite and 0.5 to 0.8 per cent. of Silicon.—These alloys grew from the outset, and reached their maximum growth in fifty to sixty heats. The bar with the lowest silicon showed the smallest, that with the highest silicon the largest growth.

Two white irons, A and M, contracted and afterwards grew. Both these irons became grey on heating, and the change from shrinkage to

growth coincided with the appearance of free carbon (temper carbon); and in the case of M as the free carbon increased, so did the growth. Free carbon is thus an indispensable factor in the growth of cast iron under these conditions.

It was shown by calculation with existing (though unsatisfactory) data that the growth of A and M observed may be accounted for by an expansion consequent on the deposition of temper carbon in accordance with the equation:

$$\text{Iron carbide } (Fe_3C) = \text{iron } (3Fe) - \text{temper carbon } (C).$$

Iron-carbon-silicon Series.—Alloys, N to S. Carbon, 3.4 to 3.98 per cent.; silicon, 1 to 6 per cent. in steps of 1 per cent. Other constituents low and constant. Growths of between 15 and 63 per cent. were found, proportional to the silicon present and far in excess of what could be caused by the conversion of combined to free carbon. In fact, they were largest in the high-silicon alloys which contained no carbon combined with iron.

These bars had been machined at haphazard, some from the gate, others from risers in different positions. The rate of growth was shown to depend on the position, bars from the gate growing more quickly than those from risers. That end of a bar nearer the gate grew more quickly than the other. The ends of some of the bars grew more quickly than the centers.

Bars cut from comparable positions were next tested (NN to SS series). They were all cut from the top of a riser, and grew more slowly than the N to S series, but ultimately reached about the same growths.

This series of alloys showed that gases played an important rôle in their growth.

The curves of rate of increase in weight were found similar to those of growth, and established an intimate connection between the two.

The final increase in weight followed the same order as the percentages of silicon, and was shown to be the resultant of three processes: (a) A partial oxidation of carbon which diminished with increase of silicon and became nil at 6 per cent.; (b) a probably complete oxidation of silicon, originally present as the silicide of iron, to a mixture of iron oxide and silica; (c) a partial oxidation of iron uncombined with silicon.

The growth of the alloys, N to S, was thus shown to be bound up with a change in the chemical condition of the silicon.

The gradual penetration of gases into the alloys during growth was studied microscopically in P, and the structural changes recorded. Large amounts of gas are dissolved at certain stages. Graphite is displaced from its original position—the spaces left are oxidized—and numerous small holes are formed. The structure is revolutionized.

IRON-SILICON ALLOYS.

Iron-silicon Alloys.—T, U and V. Silicon, 0.65 to 2.71 per cent.—containing no graphite, were shown not to grow appreciably after repeated heatings.

With regard to the influence of gases, alloy S, which grew 62 per cent. in a muffle, did not grow when heated in vacuo, but contracted (0.04 per cent.), and gave up 1.11 times its volume of gas (87.5 per cent. hydrogen and 12.5 per cent. nitrogen). Another bar, heated in vacuo until it had ceased evolving gas, was heated in a muffle and grew 67.7 per cent., showing that the growth of this alloy is entirely due to the penetration of gases. About 47 per cent. of the penetrating gases consist of oxides of carbon. They are responsible for the oxidation of the iron silicide which is so closely bound up with the growth of this alloy.

The more rapid growth of the ends than the rest of the bar is readily explained, as they expose more surface to the penetrating gases.

The more rapid growth of the end of the bar nearest the gate of the casting is connected with the fact that it contains more dissolved gas; but the cause is not yet clear.

Alloy N, which grew 13.5 per cent. in a muffle, grew 11.1 per cent. when heated to constant volume in vacuo. It gave up 1.17 times its volume of gas, 98.5 per cent. of which was hydrogen. As regards growth in vacuo this alloy thus furnished a complete contrast to S. This very close-grained iron probably does not give up all its gas on heating, some remaining permanently imprisoned. Its coefficient of expansion being about 400 times that of the iron at the surface of contact between the iron and the graphite plates, the pressure will be sufficient to produce permanent expansion.

N appears to be an alloy whose dissolved gases are capable of causing a large growth, at any rate where external gases are removed.

The growth of N in a muffle is probably caused by the combined effect of originally dissolved hydrogen and the penetrating oxides of carbon.

The research has thus disclosed, among the grey irons, alloys, *e. g.*, S, whose growth in air on heating is entirely due to oxidizing gases penetrating their interior; and also alloys, N, where this may not be the sole agent, but where originally dissolved gases contribute to some extent to the growth.

THE AGENCIES AT WORK IN THE GROWTH OF GREY CAST IRONS ON REPEATED HEATING.

Annealing ovens are grey irons, containing graphite and silicon, whose presence is intimately connected with growth; also manganese, sulphur and phosphorus, whose influence has not been investigated, and some dissolved gases, and they are exposed in practice to the direct action of flame gases. They consist of four main structural constituents, which may almost be considered as three: A solid solution of iron silicide in iron and manganese; graphite; some combined carbon—pearlite; phosphide eutectic.

With repeated heating 3 tends to pass into 2.

It has been shown that, after the first heating, gases have penetrated to a certain depth, and also that seventeen hours' heating produces no more growth than three, the actual time required depending on the size of the oven, and the rate of supply of heat for the attainment of a steady temperature, and corresponding volume. Although the gases penetrate to a certain depth during this period, possibly along slits existing between the graphite plates and the solid solution of silicide in iron, and through existing holes, they are not actually absorbed by the solid solution until the oven is cooling. During this absorption the oxides of carbon first oxidize the iron silicide at the boundaries of the crystals, this reaction being accompanied by growth and incipient disintegration. Minute cracks are formed, and possibly a series of minute explosions owing to the reaction between the dissolved hydrogen and the penetrating oxides of carbon. The mechanically weak graphite may be disintegrated and forced into holes originally existing in the metal. Nitrogen enters along with the oxides of carbon, and is absorbed to some extent (see gas analyses), the net result being a slight growth of the oven by the time it has cooled down.

INFLUENCE OF GASES.

When the oven is next heated, the furnace gases penetrate a little farther, owing to the fresh avenues opened up by the reactions just described. On cooling, these reactions are repeated, more of the iron is disintegrated, and a further growth takes place. Simultaneously, if the iron is close grained (like alloy N) hydrogen and nitrogen are liberated at the boundaries between the solid solution and the graphite. Their high coefficient of expansion causes the graphite to be disintegrated, and collect in the

holes of the casting. Where the boundary between two holes consists of graphite, it is destroyed by this action, and the two holes become one.

As the heats progress these changes continue. At each heat the external gases penetrate a little farther, and finally work right through the oven, the rate depending on its chemical composition, physical texture, etc. When this condition is reached, the oven has grown to its full extent. Some of the graphite has been burnt off, all the silicide of iron has been oxidized (probably to a mixture of iron oxide and silica), and some iron has been oxidized as well. The main cause of growth is disintegration by the oxidation of the iron silicide. In close-grained irons the pressure of dissolved gases also contributes to the growth. At the conclusion of the process the structure has been revolutionized; the oven has lost the properties of cast iron, has no mechanical srength, and can be sawn like a piece of chalk.

The authors recommend that white irons should be tried for annealing ovens instead of grey, the most suitable composition appearing to be an iron with about 3 per cent. of carbon and as few impurities as possible. Of these silicon is the most important, and should not exceed 0.2 to 0.3 per cent. This iron would probably shrink slightly on repeated heating. White irons higher in carbon will tend to deposit temper carbon, and the material will grow, though less than grey iron. White iron may prove to be unsuitable, because it may crack on heating, but this difficulty may perhaps be overcome by modifying the design.

BRASS AND OTHER COPPER ALLOYS USED IN MARINE ENGINEERING.*

By J. T. Milton, Vice-President, Inst. Mar. Eng.

The first point to be mentioned is that of nomenclature, and this is found to be a very difficult matter. Marine engineers, as well as many others who use brass, bronze and various alloys of copper and other metals, have very vague ideas as to names, calling all sorts of metals "brass" which are not true brass, and calling others "bronze" which are more nearly allied to brass than to bronze. In fact, we generally call parts of bearings "brasses" which are made of so-called gun-metal, phosphor bronze or anything but brass. Even when they are made of cast steel or cast iron and lined with white metal they are often still called "brasses." Some of the white metals are sometimes called "bronzes."

The subject of nomenclature of alloys has been dealt with comparatively recently by the Institute of Metals, which, in conjunction with other Institutes, appointed a committee to report on the matter. Their first report was made in March, 1914, and will be found in Vol. XI of the Transactions of the Institute.

Their recommendation was that the term "brass" should be restricted to alloys of copper and zinc only, but containing more copper than zinc, *i. e.*, containing over 50 per cent. of copper. An alloy containing a third metal is to be denoted with the name of the additional element used as a prefix. Thus the alloy containing 1 per cent. of tin, 29 per cent. of zinc, and 70 per cent. of copper used for Admiralty condenser tubes should be called "tin-brass." Similarly they recommended that the term "bronze" should be used to denote an alloy containing more than 50 per cent. of copper, the remainder being tin only. If other metals were present then their names should be used as prefixes, thus the ordinary Admiralty mix-

* Abstract of a Paper read by Mr. J. T. Milton (Vice President) at a meeting of the Institute of Marine Engineers on May 1.

ture usually called "gun-metal," containing 88 per cent. of copper, 10 per cent. of tin, and 2 per cent. of zinc, should be called "zinc-bronze." It may be stated that these names have not yet come into general use, and both the terms "brass" and "bronze" are being frequently used in most misleading ways

Even the name gun-metal conveys no precise idea, either of the composition of the metal or of what a gun is made of, as when copper alloys were used for making guns they were generally composed of copper and tin only; that is to say, they were "bronzes," whereas the term "gun-metal" is now more often employed to denote an alloy containing zinc in addition to copper and tin. Moreover, these guns were generally spoken of by the soldiers who used them as "brass" guns.

In this paper the alloys will be spoken of by the names usually employed, although they are admittedly not precise, but it is hoped that no misconception will arise through the use of these common names, as it is intended in all cases to state the main elements of their composition.

Copper-tin alloys were known and used in the remote past, in the so-called "bronze" age before the use of iron. It is stated that a rod of metal found by Dr. Flinders Petrie, at Meydum, was found to contain 89.8 per cent. of copper and 9.1 per cent. of tin. It was estimated to date from 3700 B.C. Objects of bronze were found at Troy, supposed to date from 1200 B.C. The first artificer we read of in the Bible was Tubalcain, who was an instructor of every artificer in brass and iron. In Hiorns' book on alloys there is a quotation from another work, in which it is stated that the word translated as "brass" in this and other places in the Bible, probably refers to "bronze," an alloy of copper and tin. From Percy's Metallurgy, however, it is evident that real brass, *i. e.*, an alloy of copper and zinc, was in use in Roman times. He gives the analyses of four ancient coins, all containing not less than 10 per cent. of zinc, of dates about 20 B.C., 60 A.D., 79 A.D., and 120 A.D., and also analyses of other ancient brass coins of somewhat later period; one of these has a composition which would be suitable for making condenser tubes. Percy states that although zinc was used in these ancient alloys, it was not known as a separate metal. He implies that it was not until the 16th century that zinc was so known, and that from the Roman times until then the brass alloy was made by what is known as the "calamine" process, a method which was in use in this country as late as 1859.

Calamine is a zinc ore consisting primarily of carbonate of zinc. In making brass, 100 pounds of calcined calamine was mixed with 40 pounds of ground coal and a little water. This was then mixed with 66 pounds of bean-shot copper and charged into crucibles which, in groups of nine, were placed in a furnace. The heat applied permitted the coal to combine with the oxygen of the ore. At a temperature somewhat below the melting point of copper the zinc which was reduced was vaporous and combined with the copper, which, being in small pieces, presented a large surface to the action of the zinc vapor. The brass produced in this way has about the composition of two parts copper to one of zinc. "Calamine" brass was for a long time thought to be of superior quality to brass produced by the direct mixture of zinc and copper, which is the method of manufacture now always adopted.

Copper and zinc alloy in all proportions, and the alloys throughout their range, have been scientifically studied perhaps more than any other series of alloys. Law gives a table of no less than 144 records collated for the Committee on Alloys appointed by the United States Board (reported in 1881). Mr. G. Charpy, an eminent French metallurgist, in a volume upon alloys, published by the Société Nationale d'Encouragement, Paris, has given a very complete account of the microstructure of these alloys. A. H. Hiorns in his book on "Metallic Alloys," gives a table of 54 different alloys mainly of copper and zinc, but a few containing small quantities of other

metals. He also gives a table of the freezing points of the alloys throughout the range, which table agrees with the curve given in the Fourth Report of the Alloys Research Committee of the Institution of Mechanical Engineers, which report deals with alloys of copper and zinc, and also those of copper and tin.

This report points out that the brass alloys which admit of industrial application are those containing from a minute proportion of zinc to 45 per cent. of that metal. When the proportion of zinc becomes as high as 50 per cent. the alloys become fragile. The report states that it is found that generally the limit of elasticity, the resistance to penetration and the hardness increase with the proportion of zinc, the tenacity also increases until it attains a maximum at about 45 per cent. of zinc. The extensibility, compressibility and reduction of sectional area also increase with increase of zinc, but attain their maximum at about 30 per cent.

In alloying copper and zinc there is no doubt whatever that true chemical union takes place between the metals, but this can only occur in atomic proportions. The atomic weights of the two metals are nearly equal, being 63.2 for copper and 65 for zinc. Various experimenters have considered that the compounds Cu Zn, Cu_2 Zn, Cu_3 Zn_2, Cu_3 Zn, all may exist. In the table of brass alloys quoted both by Hiorns and Law, compositions are given ranging by units right up to Cu_{10} Zn, and down to Cu Zn_3. What might fairly be considered is that the alloys containing the least proportion of zinc are composed of a compound of copper and zinc existing as a solid solution in copper, and that those containing more zinc as solid solutions of one zinc copper compound in another compound of different proportion.

When copper and zinc alloys are examined microscopically by suitably preparing and etching sections it is found that all those containing from 67 to 100 per cent. of copper have a uniform crytalline structure. Those of less copper content down to about 55 or 50 per cent. have a duplex structure; that is to say, they consist of two constituents, at about 50 per cent. one constituent only is present. The above refers to the metal, both in the annealed and also in the "hard worked" condition. If, however, the metal is raised to a very high temperature and quenched, the duplex structure is found to have disappeared. Apparently the two constituents mutually dissolve into one another at a high temperature, and are retained as solid solutions by quenching. Slow cooling, however, permits them to again separate, and the slower the cooling the more complete is the separation which takes place. In the alloy named "muntz metal," containing approximately 60 per cent. copper and 40 per cent. zinc, the separation is well marked.

All the alloys containing 55 per cent. and above of copper have their useful properties considerably improved by working. Those below about 66 per cent. of copper may be forged or rolled hot, muntz metal being easily so worked, above 67 per cent. it is generally considered that the metals can only be worked cold. As the critical proportion, say 66 or 67 per cent. is approached, more care is required in forging hot. Muntz metal and the other low-zinc alloys can also be worked cold as well as hot. In all cases of cold working every deformation given to the metal hardens it and renders it less able to withstand further deformation, but this hardness may be removed by annealing.

A remarkable example of how cold work can alter the shape of a mass of metal without producing fracture was given by Sir William Anderson in the discussion on the Fourth Report of the Alloys Research Committee, in an explanation of the manufacture of brass cartridge cases for 6-inch quick-firing guns. These are made from an alloy of 70 per cent. copper and 30 per cent zinc. This is rolled cold into a plate from which circular discs are cut. The discs are then pressed cold into a cup shape, then annealed and cleaned. The cups are then gradually drawn out into close-

ended tubes, thinner and thinner, each draw, except the last one, being followed by annealing and cleaning. In all there are eight annealings, the whole of the work being done cold.

The same alloy, 70 per cent. copper and 30 per cent. zinc, is also used for making condenser tubes, although sometimes a slightly higher percentage of zinc is used for this purpose, say up to 33 per cent., and similar metal is also sometimes used for cartridge cases. For the Admiralty, condenser tubes have a little tin added to the mixture, their standard proportions being 70 per cent. copper, 29 per cent. zinc, and 1 per cent. tin. In making condenser tubes the metal is first cast into tubes about 3½ inches external diameter, ½ inch thick. These tubes are then drawn cold upon smooth mandrels several stages until they become of the finished size, each draw, with the exception of the last, is followed by annealing. The last draw leaves the tube "hard." The amount of "draw" or "pinch," or reduction of cross-sectional area at the last operations, is such as to leave the tube sufficiently hard to withstand the tendency to crush, owing to its being tightly packed in the tube plates, but not so hard as to be brittle.

As a surface defect on the inside of the original cast tube would not be entirely removed by the drawing process, it is usual to bore out the cast tube before any drawing takes place, and so produce a clean and good surface to start from. This, however, is not always done.

The same alloy, viz.: 70 per cent. copper and 30 per cent. zinc, is used for making turbine blades. The process of making them, so far as the author is aware, has not been made public.

Condenser ferrules are made of an alloy containing a little more zinc than the condenser tubes. They are machined from tubes "hard" drawn, as for them hardness is a desirable quality, enabling them to be screwed up tight upon the packing without deformation.

The brass tube plates of condensers are usually made of "muntz metal," *i. e.*, an alloy of 60 per cent. copper and 40 per cent. zinc. For Admiralty work the mixture specified is 62 per cent. copper, 37 per cent. zinc, and 1 per cent. tin, and this metal is called Naval brass. The tin is required as it is thought that its presence assists in resisting corrosion. These plates are rolled hot.

It has been stated that with these brass alloys greater strength is obtained with those containing a larger proportion of zinc. Many figures are given in different books of test results of the various brasses. Their diversity is due partly to variations in the quantity of work put upon the metal, partly to the greater or less amount of annealing to which it has been subjected, and partly to the effect of impurities in the metal. Quoting the report on Alloys Research previously referred to, the alloy "with 40 per cent. of zinc elongates 40 per cent., with a strength of 23.8 tons per square inch. By adjusting the softening effect due to annealing and the strength produced by working, it is easy to increase the tenacity of bars and sheets to 38.09 tons per square inch, and to attain even greater strength in the case of wires."

In the same report there is a statement by Dr. D. Watson, the manager of the Broughton Copper Co., of Manchester, showing the effect of small quantities of impurities upon the strength and ductility of brass. He gives the following:

	Tenacity per square inch.	Elongation per cent.
Brass containing 70 per cent. of copper and 30 per cent. of zinc	20.2 tons	65½
Same brass with the addition of 0.5 per cent. of iron	24.2 tons	31¼
Same brass with the addition of 0.5 per cent. of iron and 0.1 per cent. of phosphorus	26.5 tons	28

The above samples were thoroughly worked, and afterwards annealed before testing.

Dr. Watson stated that the effect of iron was very marked, and that aluminum manganese and tin also had influence in increasing the tenacity and elastic limit, although they decreased the elongation.

It has been mentioned that when much cold work has to be put on brass, as, for instance, in the manufacture of cartridge cases, seamless tubes, etc., the brass has to be frequently annealed. The temperature at which this takes place is of great importance. Law quotes some information obtained from Mons. G. Charpy, which shows that when brass of 70/30 composition has been worked to its maximum condition of strength of over 30 tons per square inch, the full effect of annealing is produced by a temperature of 600 degrees C., which reduces the strength (cold) to 17.4 tons per square inch. Annealing below 280 degrees C. has practically no effect, at 420 degrees C. there is a marked softening effect, but the maximum is reached at the temperature stated above.

Some information as to the increase of tenacity due to cold work and also as to the effect of annealing at different temperatures was given by Captain C. Grard, of Paris, in a paper read at the Congress of the International Association for Testing Materials, held in 1912, at New York. He dealt with copper and also with brass containing 90 per cent. copper, 10 per cent. zinc, and brass containing 67 per cent. copper, 33 per cent. zinc, the latter being an alloy used for the manufacture of cartridge cases by the French War Department. He states that the brasses experimented upon were of good quality, very malleable when cold, but fragile and short when hot.

He states that the effect of cold working was the same, whether produced statically (drawing and rolling), or dynamically (under a drop hammer), and that the effect of cold work was completely removed by proper annealing. In making experiments with drastic amounts of cold rolling recourse had to be had to successive cold rollings without intermediate reheatings

To denote the amount of cold work he employs numbers represented by $\frac{S-s}{s} \times 100$, where $S =$ the original section of the specimen, and $s =$ the reduced section after the cold work. Hence, if before annealing the section is reduced to one-half the original, the cold-work number would be represented by 100.

The effects of cold working upon the tensile strength, limit of elasticity and elongation are shown in Fig. 1 for the 67/33 brass, and in Fig. 3 for the 90/10 brass, and the effects of annealing at different temperatures are shown in Figs. 2 and 4.

It will be seen that with both alloys the effect of cold working, while increasing the strength, decreases the ductility very considerably, so that the danger of cracking the metal renders it inadvisable to employ too large an amount of cold work without annealing.

It is well worth noting in Fig. 2 that whether the alloy is much or little cold worked, the effect of annealing at between 250 degrees C. and 300 degrees C. is to bring it back to one standard condition of softness.

The shape of the annealing curves in Figs. 2 and 4 show that if the metal has been rendered too hard by cold rolling it is very difficult to adjust the temperature of annealing so as to only partially soften it in cases where complete annealing would render the metal too soft. If, therefore, only a slight hardness is required for any special purpose, it is better to obtain it by adjusting the amount of cold work to give the necessary property rather than to attempt to first make it very hard, and then to soften it to the required degree.

Although valuable properties are given to these brass alloys by cold work, such as increase of strength and of elastic limit, it must not be for-

gotten that the ductility is much reduced at the same time. When brass is subjected to much cold work it is liable to spontaneous fracture. Cases are not unknown of condenser tubes spontaneously cracking longitudinally, and of cold-rolled brass rods being found split after being stored for some time. Such occurrences are sometimes called "season cracking." The

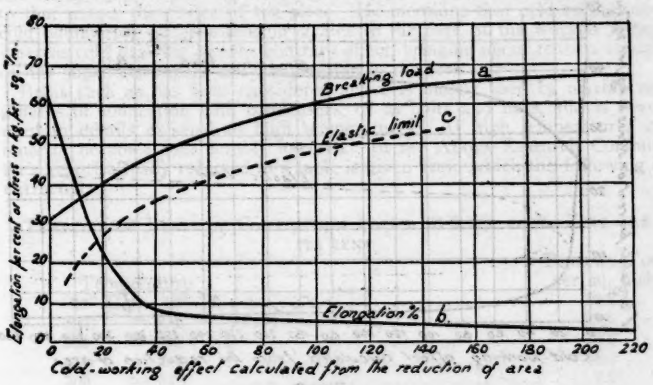


Fig. 1.

A indicates the breaking load in Kg. per sq. m/m.
B " " elongation.
C " " elastic limit.

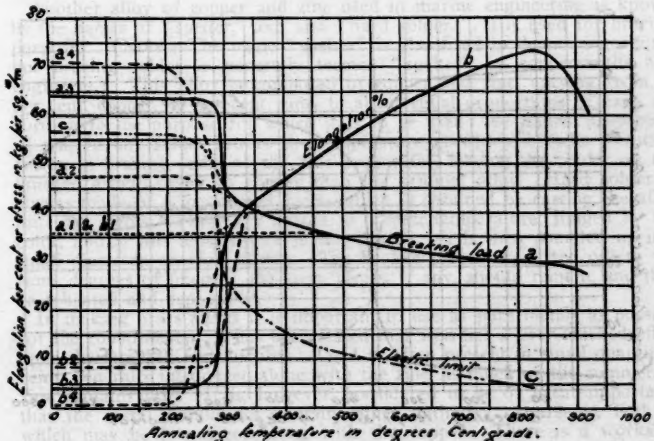


Fig. 2.

*A*1 *A* indicates the breaking load for brass cold worked to 15 per cent.
*A*2 *A* " " " " " 50 "
*A*3 *A* " " " " " 125 "
*A*4 *A* " " " " " 300 "
*B*1 *B* elongation for brass cold worked to 15 per cent.
*B*2 *B* " " " 50 "
*B*3 *B* " " " 125 "
*B*4 *B* " " " 300 "
C elastic limit.

cause is to some extent obscure, but the fact that it occurs shows how unwise it would be to rely upon what may be termed an artificial strength produced by cold rolling for articles in which a failure would cause disas-

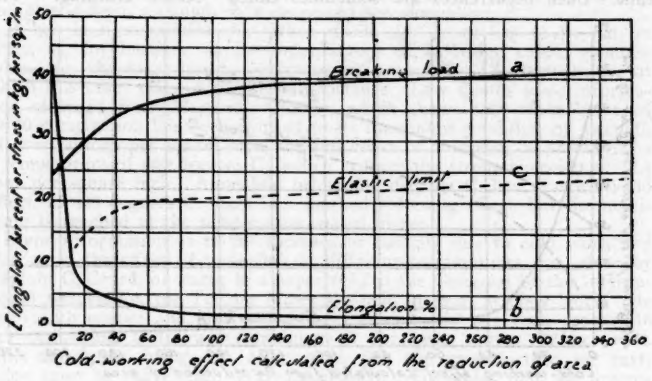

Elongation per cent or stress in kg. per sq. mm.
Breaking load
a
Elastic limit
c
Elongation %
b
0 20 40 60 80 100 120 140 160 180 200 220 240 260 280 300 320 340 360
Cold-working effect calculated from the reduction of area

Fig. 3.

A indicates breaking load
B ,, elongation.
C ,, elastic limit.

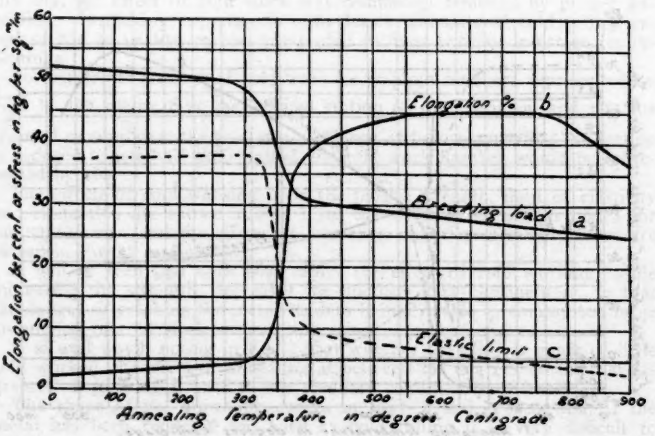


Fig. 4.

A indicates breaking load.
B ,, elongation.
C ,, elastic limit.

ter. This is sometimes not sufficiently realized by engineers who think they are securing a better article by specifying a stringent condition of high tensile strength and high elastic limit, which can only be obtained by cold rolling.

Useful information on the effects of cold work upon metals is given in the May lecture of 1914 by Professor E. Heyn, of Berlin, delivered at a meeting of the Institute of Metals, in which he shows that internal strains are set up by cold working affecting the metal near the surface differently to that nearer the center of the piece. He mentions that cold rolling and cold hammering produce tension strains in the core of the worked pieces, whereas cold drawing has the contrary effect, bringing about tension strains on the superficial layers and compression in the core.

Brass such as has been considered above is mostly used by marine engineers in connection with condensers, or as bolts and nuts, and is never used in details exposed to high temperatures. At high temperatures its tenacity becomes considerably lowered. In the Alloys Research Committee's report already referred to a table is given from which the following is extracted:

TENACITY OF BRASS OF COMPOSITION COPPER 61.2 PER CENT., ZINC 38.8 PER CENT.

Temperature.	Tenacity Tons per sq. inch.
20° C = 68° F	20.7
100° = 212°	13.6
200 = 392	12.6
250 = 482	10.4
300 = 572	7.3
350 = 662	6.8
450 = 842	3.6
500 = 932	2.8

Another alloy of copper and zinc used in marine engineering is known by the names of "spelter," and also "hard solder." It is used for brazing purposes. (Note.—The name "spelter" is also used to denote ingot zinc, but brazing solder is generally termed "spelter" by coppersmiths and engineers.) This alloy is composed of copper and zinc, varying from 60 per cent. copper, 40 per cent. zinc, to about equal proportions of each according to the purpose for which it is to be used; the higher proportion of zinc in any solder lowers the temperature at which it melts. During fusion in brazing some of the zinc evaporates, leaving the solder on the finished article richer in copper than the original alloy. Hard solder is always made of a granulated form, which is obtained by casting the alloy into flat ingots. These are heated to a high temperature, judged by the color, and at this temperature is very brittle. It is then pounded up into small pieces in an iron mortar. The broken pieces are sifted out to various degrees of fineness, the large pieces, if any, should remain, and then be reheated and repounded.

In making brass alloys it is important to use as pure metals as possible for the constituents. It has been stated that iron has a very marked effect upon the strength and ductility. Lead is often present in small quantities, being generally introduced along with the zinc, as much of the commercial zinc contains lead. It is, however, considered to be of great importance that the metal should not contain either antimony or arsenic, both of which may be introduced along with the copper. There is a workshop test known as the muntz-metal test, practiced in order to detect these impurities if their presence is suspected, either in copper or brass. A weighed portion of the copper or brass to be tested is melted in a small crucible, and sufficient pure zinc is added to bring the proportion of copper and zinc to 60/40, the composition of muntz metal. The resulting metal is poured into an open cast-iron mold, the quantity being sufficient to make a flat cake about 1 inch in thickness. This is allowed to cool. When cold it is nicked across with a chisel and broken. If the metal is good the frac-

ture is fine and silky, but if even small traces of arsenic or antimony are present the fracture is different. The skilled workman can in this way determine the proportion of either of these pernicious elements with almost the same precision as can be obtained in a chemical laboratory.

We will now consider the bronzes, *i. e.*, alloys of copper and zinc.

Pure bronzes, *i. e.*, alloys of copper and tin only, are not much used by marine engineers, as almost invariably some zinc is added to the alloys, which in their various forms are usually misnamed either "brass" or "gun-metal." These alloys are used as castings, no forging work, either cold or hot, being put upon them. The reasons for their use for many purposes are the facility with which they can be cast to the shape required, their comparative resistance to corrosive influences, and their strength.

The alloy known as Admiralty gun-metal is composed of 88 per cent. copper, 10 per cent. tin, and 2 per cent. zinc, and this is without any doubt a useful alloy. All kinds of boiler mountings, such as gage cocks, feed, blow-out and scum valves and chests, and the valves and seats of stop and safety valves, requiring a strong and incorrodible metal, are made of this alloy. Sometimes where a little harder metal is required to resist wear, such as in some valves and valve seats, water-gage cocks, etc., the zinc is omitted and pure bronze is employed of a composition of 90 copper and 10 tin.

The addition of the zinc serves a very useful purpose in rendering the casting sound, besides the influence it has on the strength of the metal. When copper is melted and exposed to the air it rapidly oxidizes, and molten copper has the property of dissolving the copper oxide so formed. When a bronze is melted both the copper and tin composing it are oxidized. The tin oxide is not soluble in the molten metal, but the copper oxide is; the tin oxide formed therefore remains on the molten metal as a scum, but the copper oxide disappears by going into solution in the molten metal. During the solidification of the metal at least part of the oxide becomes thrown out of solution and separated, and the result is a multitude of minute particles of copper oxide dispersed through the metal, which in consequence is more or less porous and unsound. If, however, there is a small proportion of zinc present, this metal has a greater affinity for oxygen than copper, and the copper oxide becomes reduced to metallic copper, giving up its oxygen to form zinc oxide, which, not being soluble in the fluid metal, rises to the surface in the scum. A small addition of zinc, therefore, permits a sounder casting to be produced.

A similar result may be obtained by the addition of a small proportion of phosphorus, which, as is well known, has a very great affinity for oxygen. When this is added to a copper-tin alloy the result is termed a "phosphor bronze." The phosphorus is added either in the form of a phosphor-copper or phosphor-tin. Phosphor-copper is commercially obtainable in two forms, one containing 15 per cent. of phosphorus, the other 10 per cent., the latter melting at a lower temperature than the former.

If the precise proportion of phosphorus necessary to remove the oxygen is added to the bronze, the metal remaining is still called "phosphor-bronze," although it will contain no phosphorus, but its properties are considerably better than those of a bronze not deoxidized.

Zinc is often used in the so-called "gun-metals" employed in marine engineering in considerably greater proportions than the 2 per cent. specified in the Admiralty mixture. It is sometimes said that the greater proportions are used because zinc is cheaper than either copper or tin. Lead is also often added, but not always with the view of making a cheaper mixture, as a reference to Mr. Dewrance's research, to be presently referred to, will show. The addition of lead renders the metal less tough, and therefore more easy to machine, and for this reason it is, to say the least, not objected to by the machinist. It is also claimed for certain admixtures of lead that they render the metal more suitable for bearing purposes, by

reducing the friction under heavy loads, and, especially in America, alloys containing considerable quantities of lead are used for bearings in heavy machinery.

The microscopic structure of the so-called "gun-metal" used in marine engineering work is peculiar. When ordinary castings are being machined with a fairly rough cut the surface is seen to present a mosaic of ordinary yellow and of a rather lighter color. Each portion showing one color is a crystal with different orientation to its neighbors, and the crystals so shown are fairly large, their size depending upon the rate at which the casting cooled from its liquid condition. When a cut surface of the metal is carefully polished and suitably etched and examined under a microscope, it is seen that each of the above-mentioned crystals is really composed of a number of "dendrites," arranged in regular geometrical fashion like the divisions of a fern leaf. What happens in the cooling out of the metal is that the particles which first solidify are richer in copper than the average of the metal. They arrange themselves in a skeleton form of dendrites. Next to solidify are other particles not quite so rich in copper as the preceding; these arrange themselves outside those already formed, and the process goes on, the skeleton dendrites becoming more and more clothed with metal less and less rich in copper, and the portion of the metal which still remains fluid becomes relatively poorer in copper and richer in the tin or tin and zinc, until a point is reached at which it all becomes solid. The dendrites thus formed have been referred to by American metallographists as being of "onion" type, not because the shape is that of onions, but because it is formed of successive layers deposited on one another. Suitable etching of sections of the dendrites shows by the deeper colors of the centers that this difference of composition exists.

Where the copper is not more than about 91 per cent. of the composition the metal cannot well be worked either by hammer or by roll, either hot or cold.

With greater proportion of copper than 91, especially when the metal has been deoxidized by phosphorus, some amount of work can be put upon it, and we have made in this way the so-called "phosphor bronze" rolled sheets which are used for air-pump valves, and which are hard and elastic. It is also drawn into wire for electrical purposes where high conductivity, combined with high strength and resistance to corrosion, are required.

In view of the peculiar onion-like, minute structure of gun-metal, it may be thought that it will be possible to modify its qualities by such heat treatment as will make the constituents diffuse into one another and become a homogeneous whole instead of remaining of diverse composition. Experiments in this direction were made by Messrs. H. S. Primrose and J. S. G. Primrose, and were communicated to the Institute of Metals in 1913. They experimented on metal cast in ordinary dry-sand molds, and also on samples cast in chill. Their tests included raising the metals to different high temperatures, keeping them at these temperatures for different lengths of time, and cooling some by quenching, and allowing others to cool out as in ordinary annealing. Their conclusions were:

1. No improvement results from heating and quenching, which lowers the strength of the material.

2. Simple annealing of the metals for 30 minutes very considerably increases its strength and elongation, the maximum results being obtained after annealing at 700 degrees C.

3. The homogeneity and other physical properties of the metal are correspondingly improved, but particularly the capability of the castings to withstand hydraulic pressure.

In the results quoted in the paper the unannealed dry-sand casting had a tensile strength of 17.2 tons per square inch, and an extension of 24 per cent., whilst that which had been annealed for 30 minutes at 700 degrees C. had a tensile strength of 18 tons, and an extension of 37.5 per cent. With the "chilled" castings the improvement was still greater.

It may be stated that the requirements of the Admiralty specification for gun-metal are a minimum tensile strength of 14 tons per square inch, and extension of at least 7½ per cent. in a length of 2 inches.

Although "gun-metal" is extensively used for boiler fittings, and therefore exposed to temperatures as high as that of high-pressure steam, very little information is available upon the subject of the effect of high temperatures upon the strength and ductility of the metal. There has, however, long been an uneasy suspicion that high temperatures considerably depreciate its qualities.

Mr. John Dewrance, in a paper communicated to the Institute of Metals, gives some valuable information on this subject, which shows that the suspicion was well founded. Good Admiralty mixture with a tensile strength of over 16 tons per square inch, and extension of 11 per cent. when cold, retains its strength up to a temperature of 350 degrees F. (corresponding to a boiler pressure of 120 pounds per square inch). At 375 degrees F. (corresponding to 170 pounds pressure), its strength fell to 13.5 tons per square inch, whilst at 400 degrees F. (corresponding to 245 pounds pressure), its strength fell to 9.2 tons per square inch, and its extension fell to 1 per cent. At higher temperature the deterioration was still worse. Mr. Dewrance, however, found that the admixture of a little lead to the metal considerably improved matters. If the tin and zinc were retained at 10.2 per cent., respectively, and the copper was reduced by ½ per cent., and ½ per cent. of lead added, the extension when cold was a little lowered, but the tensile strength was not affected. At high temperatures, however, a great difference was found. No reduction of strength took place until a temperature of 550 degrees F. was exceeded, and the extension was continuously improved right up to this temperature. Mr. Dewrance made similar experiments with still larger proportions of lead. His conclusions were that ½ per cent. lead gave the best results, but even more than this still left the metal better at high temperature than that containing none.

Attention will now be directed to some of the alloys for which very special qualities of strength, etc., are claimed. They are all composed mainly of copper and zinc, and all or nearly all contain a proportion of iron, some of them contain one or more other components, such as tin, aluminum, manganese or nickel. They are mainly proprietary articles, and their exact composition and the methods of making them are supposed to be trade secrets, their composition, however, can easily be determined by analysis.

The fact that iron could be added to brass, and that its addition strengthened the alloy has been known for very many years, and such metals have long been known under the name of "Sterro metal" and "Aich's metal." Law states that these two metals each contain 60 per cent. of copper, 38 per cent. of zinc, and 1.5 to 2 per cent. of iron, although the composition, so far as iron is concerned, varies within wide limits. This variation was due to the imperfect methods of introducing the iron. In 1883, Mr. Dick took out a patent for manufacturing an iron-brass, which he called "Delta" metal, by introducing the iron in the form of an alloy of iron and zinc, which could be obtained of reliable composition; other metals, however, were added. Law gives the average composition of Delta metal as copper 55 per cent., zinc 42 per cent., iron 1 to 2 per cent., with small quantities of manganese, aluminum, tin, and sometimes also lead.

Delta metal, like most of these iron-brasses, can be easily cast, and it is also capable of being rolled or drawn cold, and of being forged hot. It is also often "extruded." In this process, instead of being hot rolled from a billet, finished bars are made by forcing the highly heated metal through dies by applying a great pressure by means of an hydraulic press. By such means bars of almost any desired cross section may be made, but it is not generally considered that the quality of bar so made is equal to that of bars of the same composition made by the ordinary rolling process.

Extrusion may be applied to most of the alloys containing from 55 to, say, 60 per cent. copper.

"Manganese bronze" is another of the alloys frequently used. It really ought not to be called bronze, but might, perhaps, claim to be called manganese-brass, although the amount of manganese in it is exceedingly small. This metal is used for castings, especially for propeller blades, and it is also forged when required and rolled into bars from which studs, bolts, valve spindles, etc., are made. It is made by several firms who all have their own ways of introducing the iron, the vital constituent, into it. The basis of the iron in the case of this metal is ferro-manganese.

It has been stated that copper when melted dissolves copper oxide, which is an undesirable constituent. The manganese has great affinity for oxygen, and hence acts as a deoxydizer. It, therefore, sometimes happens that the whole of the manganese added in the ferro-manganese leaves the metal as manganese oxide; as, however, the iron remains in the metal the desired strengthening quality is preserved. Ferro-manganese also contains carbon, which no doubt assists in the deoxydizing progress. Some tin is usually added to the manganese bronze used for propeller blades, and often some aluminum as well.

Analysis shows that the iron is not always uniformly distributed through a large casting, the difference sometimes ranging to from 1.5 per cent. of iron to as much as 4 per cent., even when taken from contiguous parts of the casting. When a sample containing much iron is polished for microscopic examination minute hard spots are seen in it. These are thought by Mr. Dewrance and Mr. Arnold Philip, the Admiralty chemist at Portsmouth, to be particles of iron in alloy, due to the iron alloying with the tin, and a greater proportion being present than can be kept in solid solution.

Many analyses of manganese bronze have been published, Law gives the following:

	Copper.	Zinc.	Tin.	Lead.	Manganese.	Iron.	Aluminum.
Cast	57.20	40.14	1.18	0.02	0.03	1.33	0.10
Rolled ...	62.45	36.00	0.63	0.06	0.02	0.84	....

It will be seen how little manganese there is in either of these alloys.

Law states that nickel is sometimes added to manganese bronze used for propellers, in order to prevent erosion. It is, undoubtedly, present in one alloy used largely for forging purposes, which, however, is not called manganese bronze by its maker.

The tensile strength and elongation of good rolled manganese bronze is about the same as those of ordinary mild steel. Like mild steel when the bars are used for studs, the material ought to be annealed in order to remove from it any possible effect of cold working. When bolts are made from it by forging, the heating in the process effectually removes this particular trouble, but if bolts are machined out from the solid bar, as they sometimes are, then the bars should be annealed, as in the case of studs.

Aluminum brass, or, as it is usually called, aluminum bronze, has sometimes been used for propellers. This is very similar to the manganese bronzes, and like them contains a small quantity of iron. The aluminum is added to replace a small part of the zinc, the copper contents not being much varied.

Cast manganese bronzes of the special types as made by various makers can be relied upon to have a tensile strength of from about 30 to 35 tons per square inch. They are, therefore, about as strong as cast steel, but they do not corrode so much as steel, and can be made with good surfaces. The influence of a smooth surface in improving the efficiency of

propellers is now fully appreciated, especially in fast ships, and with high speeds of rotation. It need not be mentioned how much greater efficiency is claimed by makers of special propeller bronzes over ordinary iron or steel screws.

When so-called "bronze" propellers were first used it was found that the iron and ships suffered from corrosion in their vicinity; if by chance the paint became injured, this was put down to galvanic action. This led to a different composition being used for the propellers. Now, more trouble appears to be met with in the corrosion of the propeller itself, and in some few cases very serious deterioration has occurred, especially in quick-running propellers, such as are used with turbine engines. To meet this makers have in some cases again changed the mixtures, and "bronzes" are made which are claimed to be non-corrodible.

For yachts and small craft, which are built of wood and are copper sheathed, and in which iron or steel shafts are peculiarly liable to corrosion, forged "bronze" shafts of different sorts have been used, but it cannot be said with marked success, as all of the forgable bronzes are copper-zinc alloys, and all seem to be peculiarly liable to a form of deterioration to which the term "dezincification" has been applied. Copper-tin alloys would probably be more durable, but they do not possess the strength of the copper-zinc alloys. It is with the view of preventing, or, at least, retarding this dezincification that the Admiralty require at least 1 per cent. of tin to be added to condenser-tube metal, and to all brass exposed to the action of sea water. The question of the best metal for making screw shafts for wood vessels will soon be raised in view of the decision to build a large number of wood steamers in America.

DISCUSSION.

Sir Gerard Muntz, in a communication, said that the reason for the cracking or splitting of brass rods in store, which the author said was rather obscure, was not obscure to him. It was due primarily to irregularity of the mixture. Metal containing 1 per cent. of tin should in no case contain more than 59½ per cent. of copper. If it contained 60 per cent. of copper it should not contain more than ⅝ per cent. of tin, otherwise it passed from the alpha-beta phase to the alpha-beta-delta phase, which had a brittle constituent, hard and liable to cause disruption in the metal. The amount of work in brass rods in rolling or drawing was also a considerable element in the matter. If too much work was put on with the outside surface in a state of tension and the inside in a state of compression, it was easy to understand how the surface of the rods could be so compressed that the strain on the interior might nearly reach 23 tons, and a slight reduction in temperature and consequent contraction carried the strain past the breaking point and the rod would at once fly.

Mr. W. G. Brown said the longitudinal splitting of condenser tubes was a common occurrence and happened when the tubes were lying in a rack. He hardly thought the split in the tube shown by the author had elongated itself out of a small speck. He had never before seen a case of brass rods splitting like that shown by the author, and he did not think Sir Gerard Muntz's explanation was a really satisfactory one. He had seen tubes crack longitudinally and often crosswise in forging. With regard to the metal bolts that had been shown by the author, it was quite common for them to go in the supposed strongest place, not in the threads, but generally under the heads or in the body.

Mr. Milton said that Mr. Brown's statement as to tubes splitting in store was in direct opposition to Sir Gerard Muntz's statement. The latter said that 1 per cent. was too much tin if there was 60 per cent. of copper, but that amount of tin was in all navy condenser tubes, and in the navy split condenser tubes were very rare.

Mr. Brown said the tubes he had referred to were the common variety—70/30 and no tin. Therefore the presence of tin was not the cause of the splitting.

Mr. R. Balfour said the importance of paying attention to the brass bolts used for sea connections could not be too much emphasized. He thought mild steel might be substituted and did not understand why the use of yellow metal for this purpose was continued. Sometimes sea connections were covered with granite mixed with cement, but this formed cavities and would not adhere. Compared with olden times, sea connections were now very accessible, and they should be thoroughly exposed in order to facilitate examination of the bolts.

Mr. F. O. Beckett said that "Sterro" metal was first introduced in Vienna a few years ago by Baron de Rosthorn. It was of copper-zinc tin. The ultimate tenacity varied from 83,000 to 85,000 pounds per square inch, or an average of 80,480 pounds per square inch. It required a strain of 30,000 pounds to produce a permanent elongation of 0.002 inch, whilst the ultimate elongation with the average strain mentioned was 0.0675 inch per inch of length, whereas bronze, which began to yield at 15,000 pounds, and was fractured at 33,000 pounds per square inch, showed an ultimate elongation of 0.290 inch per inch of length. With a mixture of copper, 57.63; spelter, 40.22; iron, 1.86, and tin, 0.15, the breaking strain was 28 tons after simple fusion, 32 tons average red hot, and 37 tons drawn cold and reduced from 100 to 77 transverse central area. With a mixture of copper, 60; zinc, 34; iron, 4, and tin, 2, the breaking strain was 31.61 tons to 32.52 tons when cast in an iron mold and forged red hot, whilst with a mixture of copper, 60; zinc, 35; iron, 3, and tin, 2, the breaking strain was 38 tons with the same treatment. He did not recommend making stays for boilers of bronze, as he had had a very sad experience with locomotives due to the perishing of fire-box tubes a few years ago. Castings of Delta metal, which was a high-class bronze somewhat similar to manganese bronze, but rather harder and of a higher tensile strength, had an ultimate strength of 34 tons and resisted torsion better than bronzes, which were disappointing in this respect.

Mr. A. E. Sharp said such a paper had been wanted for some time. They had been calling all sorts of things bronze. In connection with the season cracking, he had had two or three experiences recently in connection with the re-tipping of propeller blades. They were all right when put on the ship and stowed with a strap across them, but after a time—perhaps a month—the chief engineer reported that they had decided cracks in them where they had been re-tipped. The re-tipping was done in the usual way by burning the pieces on. One peculiarity was that the cracking got worse. It began in a very small way. When the ship came back he thought it might have been an imperfect job, but the blades had been carefully examined like all the other blades, and re-tipping was not a new thing, and he could not account for the trouble commencing recently. It seemed to bear out the expression of opinion in the paper that there was something in connection with those metals that they had not got to the bottom of yet. He had also two or three experiences of corrosion of copper pipes. It was evident the builder must have had some method of fixing up the pipe different from other builders. He had had experience some years ago of condenser tubes developing very small holes, and the manager of a copper works to whom he submitted them found the holes were due to small particles of iron in the tubes. One could quite understand after reading Mr. Milton's paper that it was quite possible for small particles of iron to be present, and of course it was immediately attacked by the salt water. He had three or four of the manganese bronzes by different makers analyzed and was woefully disappointed. In some there did not appear to be a trace of manganese.

Mr. Wesley Lambert (representing J. Stone & Co.) said the reference to propellers prompted him to come forward. With regard to correct nomenclature of alloys, he thought that Mr. Milton was one of the gentlemen in a position to enforce that sort of thing, as he was in an official position. Stone & Co. had an enormous number of specifications from the various naval departments, the Board of Trade, and Agents for the Colonies, and they all had their own specifications. The Engineering Standards Committee, the Alloys Research Committee, and such institutions as the Mechanical Engineers, Civil Engineers and Institute of Marine Engineers, and other institutions, were almost in a position to enforce correct nomenclature for alloys, and he would be pleased to see that sort of thing come along. In the letter from a French engineer quoted in the paper, numbers were employed to denote the amount of cold work. He presumed that the section spoken of was the cross-sectional area. Many years ago Sir Joseph Whitworth introduced the unit of work which he described as being that amount of work which would reduce the cross-sectional area by a half, or in other words double the length. He was sorry that that had not been really carried out He was doubtful whether any engineer could give a definition of a unit of work so far as a forging was concerned. There was nothing to describe the amount of work which should be put on a bloom or a billet to produce a certain forging, and he had never seen a specification of an engineering firm in which they described the original size of the billet from which the forging was to be made. Government specifications for ordnance work had included that there should be so many units of work put on the material. That was one of the only cases in which he had seen an attempt to describe a unit of work or give any expression of the amount of work.

Mr. Milton said the Engineering Standards Specifications of forging for shipping purposes described it. It said the amount by which the forging must be reduced from the billet must not be more than one-fifth—it was a reduction of one-fifth of the body in the forging.

Mr. Lambert said he thought engineers had been remiss in not stating the amount of work that should be put on. He took it that the round brass bar 2⅛ inches in diameter referred to by Mr. Milton, and specified to have certain tests (30/20), was not a cast bar. He did not know whether it was rolled or extruded or whether it had been through the drawbench. They only knew it was there and it was not apparently a good specimen of a brass bar. He would like to know the history of it; then he might offer some suggestions. He was not altogether wedded to the idea of season cracking. It was not an unusual thing for a billet of steel of large dimensions (even 30 inches in diameter) to fracture, and not due to cold work because none was done on it. The mere effect of the variations in the temperature appeared to cause internal stresses to reveal themselves and the thing had broken in two. Gun tubes, fortunately, very rarely split in the same manner as condenser tubes had done. Mr. Milton said: "When so-called bronze propellers were first used it was found that the iron of the ships suffered from corrosion." He thought Mr. Milton meant erosion. The paper described a method of extruding billet metal, and stated that "In this process, instead of being hot rolled from a billet, finished bars are made by forcing the highly heated metal through dies by applying a great pressure by means of a hydraulic press. By such means bars of almost any desired cross-section may be made, but it is not generally considered that the quality of bar so made is equal to that of bars of the same composition made by the ordinary rolling process." Perhaps there was a good reason for the assumption, but there were hundreds and hundreds of tons of extruded rods being used and accepted by the government without demur. Mr. Milton said he had not had an opportunity to discuss or enlarge upon the point of failure of billets said to have

been made from the extrusion process, but he (Mr. Lambert) thought in one or two of these instances there was a certain amount of heat work put on after the material was made, and in the ordinary smith's shops great damage could be done to the material and sometimes failures resulted from the ignorance of the smith rather than faulty metal initially. The basis of the iron in the case of manganese bronze was ferro-manganese. In the majority of analyses the iron was there in a much larger proportion than the manganese. So-called pure manganese contained about 80 per cent. of manganese and a considerable amount of carbon. Iron must be added in addition to that carried by the ferro-manganese. A speaker was disappointed at the small quantity of manganese he found in manganese bronze. It could be manganese bronze without much manganese being in it, just as a phosphor bronze could be a phosphor bronze without any phosphor in it. As Mr. Milton admitted, it could be called phosphor bronze because phosphor had been used as the cleansing agent. There were recognized brands of manganese bronze on the market which were very constant in their content of manganese, considerably more than that shown in the analyses in the paper. He agreed that bolts machined out of the solid bar should in some instances be annealed. Metals had certain internal stresses, and in machining these, especially where there were different cross-sectional areas for different parts, it was necessary to anneal them. The Admiralty practice to a large extent was to burn on gun-metal tips to manganese bronze propellers. If manganese bronze blades cracked which were in the beta condition it was obviously wrong to burn on a tip of an alpha-beta character, because the physical properties of the two metals were quite different. When burning on a tip the blade should be heated uniformly in the neighborhood where the burn was to take place, and subsequently, after the burn had been made, heating should again be done uniformly so as to relieve all the unequal stresses which would be likely to be set up. Mr Sharp was fortunate in being immune from the trouble so long. Other shipowners had experienced it over a long period. Mr. Sharp's mention that the blade was out of pitch after the burning was performed pointed to the conclusion that it was not properly performed.

Mr. Milton said erosion meant wearing away by rubbing. In the case of the propellers, he thought there must be corrosion and that the corroded portion got rubbed off. In the case of erosion of guns there was a great temperature as well as the rushing past of the projectile. That was different from the erosion of propeller blades, which were pitted. It must in the latter case have been oxidation or something of that kind. That which was rubbed off was metal that was corroded. It was oxidized or combined with something. He thought the roughness produced a partial vacuum in the water so that the oxygen in the water came out in solution as gaseous oxygen and that that was the cause of the corrosion.

Mr. B. P. Fielden said he would be glad if the author could give any advice as to getting over the trouble of corrosion of condenser tubes.

Mr. Milton said he hoped a paper would be brought forward on that subject. The Institute of Metals and the Corrosion Research Committee had made three voluminous reports, but the subject was still under investigation. Experimental work was also being done both at the college and the Imperial Institute, and at Brighton. The subject was a very serious one. In the Royal Navy they seemed to have got over the trouble in the last two or three years. Probably they would communicate their experience in due time to the Institute of Metals. In his time they had their tubes made of the 70/29/1 composition, and were careful not to get impurities in the tubes, and they were always careful to get a certain amount of pinch or draw in the tube to make it stiff to stand the packing.—"Shipbuilding and Shipping Record."

COBALTCROM, THE NEW STEEL ALLOY.

Widespread interest has been aroused by the announcement on behalf of Messrs. Darwin and Milner, of Sheffield, of the discovery of a tool steel stated to be equal in durability and hardness to high-speed steel, yet of which tungsten is not a component. "Cobaltcrom," as the new steel is called, is the outcome of much experiment and labor, extending over many years, on the part of the head of the firm of Darwin and Milner. He had long been experimenting in the direction of finding a steel which could be made without tungsten, but would possess the necessary hardness of the so-called "high-speed" steels. About twelve months ago he discovered that by adding cobalt to chromium-carbon steel he converted a steel which had no appreciable red-cutting hardness into one which had this very valuable qualification.

It is a fairly well known fact that while tungsten high-speed steel has for some years past been extensively used for turning, planing and slotting tools, it has not been used so largely for milling cutters, taps, reamers, etc. It is estimated by Messrs. Darwin and Milner that of all the milling cutters, etc., used in the engineering world, at least 90 per cent. are still being manufactured from carbon steel, whereas, on the other hand, the reverse is the case with turning tools, nine-tenths of which are made from tungsten high-speed steel. In the opinion of the firm the reason that high-speed steel has not been applied largely for milling cutters, taps, etc., lies in the difficulty which the toolmaker experiences in hardening such tools having fine edges at the high heat required to secure from tungsten high-speed steel its best cutting efficiency. Tungsten high-speed steel requires hardening at from 1,250 degrees to 1,350 degrees C. if the best results are desired. To try and obtain this heat for milling cutters is very risky, and to harden at a lower heat gives less satisfactory results.

When the war broke out Sheffield firms received instructions that they must make milling cutters and taps from the very best high-speed steel, but there was endless trouble through the hardening process alone, with the result that carbon steel was soon substituted. Now it has been found that the maximum heat necessary for the hardening of the new steel is only 1,000 degrees C. As a matter of fact, if this heat is exceeded the tools give adverse results. This temperature allows of the manufacture of fine-edge tools of a perfectly satisfactory character with the red-cutting hardness of the tungsten high-speed steel. Hardening in practically all applications is stated to be satisfactory when the tool is allowed to cool naturally in still air free from draughts and currents, whereas with the tungsten high-speed steel, hardening from the extreme high heat renders it necessary to quench either with a strong air blast or a sudden immersion in liquid. Messrs. Darwin and Milner state that with the new steel it is possible to get absolutely the same standard of hardness throughout.

Samples of the steel have been tested in a large number of important works throughout the country, and have shown that the cobaltcrom is specially suitable for milling cutters, twist drills, reamers, taps, automatic forming tools, screw cutting, finishing tools generally, for machining gunmetal and yellow metals, as well as for high-endurance drawing and blanking dies, lathe centers, shear blades, aeroplane and automobile combustion-engine valves, and every description of gages and instruments which have to resist abrasion. A test has been made showing that with a carbon steel blade saws can cut for half a day, with a tungsten high-speed steel blade for two days, but with the new steel four days without the need of grinding. We are informed also that particulars have come to hand of tests made in America between the new steel and the Becker iridium-cobalt-tungsten high-speed steel, these proving the tungstenless steel to be the equal of the famous American product.

Messrs. Darwin and Milner have applied for patent rights the world over, including, by permission of the Patent Office authorities, Germany and Austria. A point which should not be overlooked is that the cutting efficiency of the new steel is quite equal, even in the form of castings, to that of tools made from the forged or rolled bar in which high-speed steel is supplied commercially, and as the material in the molten state is much more liquid than high-speed steel, it lends itself to all forms of tool-form castings. We are informed that the makers are preparing to put into operation an organization to supply orders for milling cutter blanks in this new steel within 48 hours, and this should be of great assistance to engineering concerns in times like the present.—"Engineering."

CONDENSER TUBE CORROSION

By William Ramsay, F.I.C.

In the engineering profession the cause and prevention of condenser-tube corrosion is one of the most burning questions of the day. Large sums of money have been devoted publicly and privately to investigate the difficulty, but so far little, if anything, has been accomplished to the end in view.

The work of the Corrosion Committee of the Institute of Metals was looked forward to with great expectation, but it is to be regretted that the results published in the various reports are, so far as they go, of academic interest rather than of practical engineering value. While throwing some light on the mechanics of brass corrosion in general, from an engineering standpoint they only indicate, with close agreement to practice, the order of durability of tubes made from the various alloys in general use. Had the work taken a different course, one cannot but think that something of more immediate value to engineers might have been forthcoming.

Retubing a condenser is always a costly business, even after the tubes have done good service for what may be regarded as the natural period of their life, not uncommonly ten, fifteen or even twenty years; but unfortunately one is frequently faced with cases where they have broken down after a comparatively short period of use, often only a few weeks or months. This is a much more serious matter, and it is in just such cases as these where the Corrosion Committee has failed us. Again, unfortunately, this type of defect has been increasingly familiar during the past ten or fifteen years, a fact which has given rise to the opinion that the extra purity of the metals now employed is responsible for the trouble. But, on looking up analyses of condenser tubes over a period of twenty years, I find that there is no justification for such a suggestion. The alloys now used are, so far as purity is concerned, probably no better or worse than the older material.

Having been interested in this question for many years, and being exceptionally well situated for the study of the subject, it has been suggested that an account of my observations and some experimental results might add to the not very large total of our knowledge of this elusive problem.

When a breakdown occurs it is usual to hand a piece of the defective tube to the chemist for analysis and explanation. The analysis is a simple matter, but since the corroded metal has disappeared and the tube adjacent to the corroded parts is, as a rule, normal, the explanation is more difficult. Again, analysis of the corrosion product leads to no conclusive result; it generally contains the metals of the alloy as basic chlorides, carbonates, oxides or even sulphates, together with adventitious matter such as oxide

of iron and organic matter; and as much of this product is washed away in both the soluble and insoluble form, an examination of the residue very seldom points to anything useful.

From time to time I made a number of experiments in order to throw some light on the cause of corrosion; many were abortive, and in some cases the true significance of the results was not always immediately apparent, or was even contrary to what was expected.

The chief difficulty of experiments of this nature lies in the impossibility of reproducing actual working conditions. Simple immersion tests may afford information on certain points, but can rarely indicate the cause of failure. This also applies to model condenser arrangements, in which the same water is circulated many thousands of times. Such water may lose its corrosive properties, or, on the other hand, it is conceivable that it may even become more corrosive with continued use, so that results obtained by these methods must be accepted with certain reservations. One of my first experiments was with a condenser tube smaller than, but similar to, that used by the Corrosion Committee in their work, but after a few months this was abandoned, as it did not lead to conclusive results, and I do not propose to refer further to it. Other experiments will be mentioned in their proper sequence.

The nature of brass corrosion is, at the present moment, fairly well understood, and may shortly be described as being mainly, if not entirely, electrolytic in character. It is not my purpose to go into the micro-metallurgy of the various alloys used in the manufacture of tubes; it is sufficient to say that the commercial brasses are not homogeneous bodies, but are a kind of conglomerate, the various constituents of which have different solution pressures or electrolytic potentials. Even the 70–30 brass, which should theoretically have only one constituent, is not entirely homogeneous, but shows decided traces of different material at the junctions of the crystals; but even assuming that it is homogeneous, as will be shown later on, it soon acquires a conditon which renders it "autocorrosive"; by this term I mean that brass possesses within itself, or can acquire, the positive and negative elements necessary to form a galvanic couple, which, in sea water, results in corrosion.

Corrosion of a condenser tube may be said to commence from the moment it is put into use, and the extent to which it proceeds depends upon such factors as composition, time, temperature, water, and so on. The influence of these factors has been very ably investigated by Dr. Bengough and his co-workers, and published in their reports. I do not think that different sea waters can have a varying influence on the rate of corrosion, but estuary waters, containing sewage, sand, etc., certainly have such an effect.

The effect of corrosion in condenser tubes is varied. Copper and zinc may be removed uniformly from the surface, so that the walls become thinner and preserve the yellow color of the brass, or, what is more commonly the case, zinc is dissolved out, leaving the copper or a copper-rich material as a coherent film, the original dimensions of the tube being nearly preserved. This action has been labeled with the rather cumbersome name of "dezincification," a term which has led to the idea that the solution of the zinc is chemical rather than electrolytic, but it is purely electrolytic. In the case of the 60–40 tubes the action takes place between the alpha and the beta constituents and may proceed to completion, the tube retaining its original dimensions and, to the naked eye, presenting the appearance of a solid copper tube. On microscopical examination, however, it is seen to consist of coppery material preserving the form of the alpha constituent, the space previously occupied by the beta being more or less filled by corrosion product, or possibly quite empty. I have in my possession a piece of 60–40 tube which has undergone this complete de-

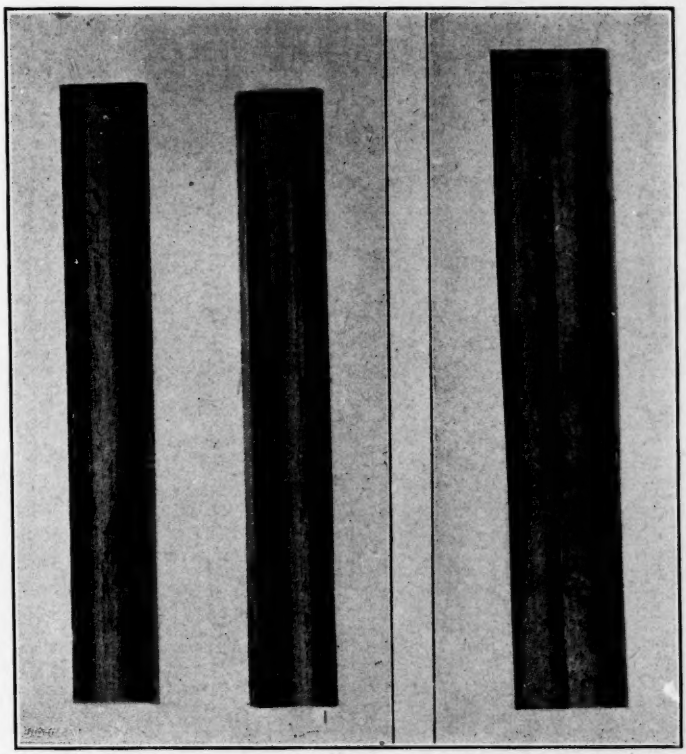


FIG. 1. FIG. 2.

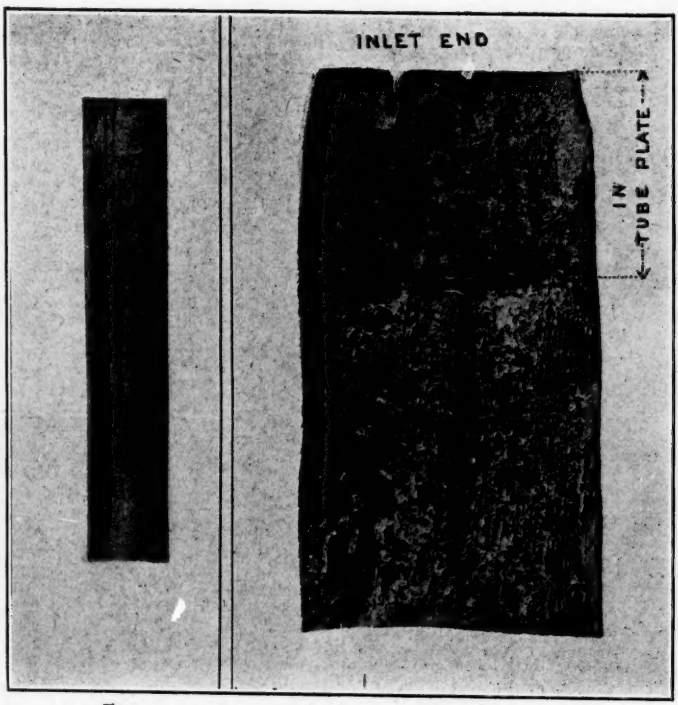


FIG. 3.

FIG. 4.

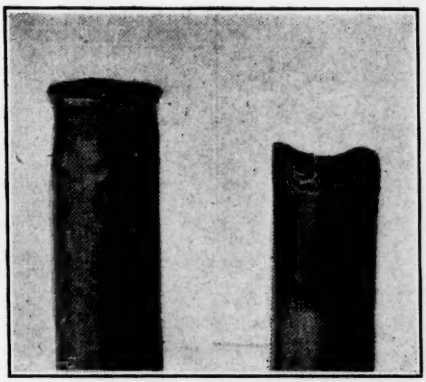


FIG. 5.

Fig.6.

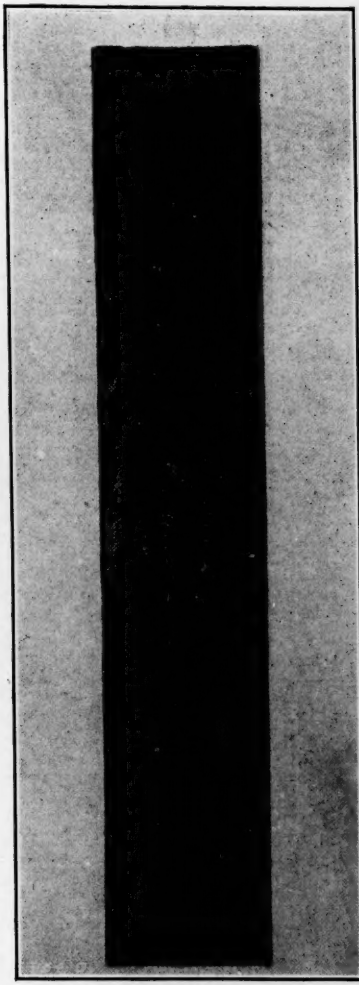


FIG. 7.

zincification; the information which accompanied it was to the effect that it broke into short pieces when drawn from the condenser, and was so soft that it could be distorted between the finger and thumb, but on drying it became quite hard again. Apparently it was only held together by the interlocking skeleton of the alpha brass.

With 70-30 tubes and Admiralty alloy the action is probably somewhat different. Owing to some slight inequality in the composition—or, possibly, to a thin film of copper oxide, from which the tubes are rarely free, and which in contact with salt water is reduced to copper—the equilibrium of the brass becomes upset, and a more or less continuous film of copper results; electrolysis then takes place between the copper and the underlying or adjacent brass, resulting in a gradual concentration of copper. With these tubes dezincification is as a rule superficial, the more compact nature of the residual copper sealing off the metal underneath from further action. That the Admiralty alloy and the newer lead brass are more resistant is probably due to the more insoluble corrosion product being impacted in the porous copper residue.

But dezincification is not the simple process outlined above. If some of the copper goes into solution—and there is ample evidence that it does—it may be redeposited as metallic copper on the brass, an equivalent amount of zinc going into solution. I have proved that cuprous and cupric chlorides, oxide of copper, and, in fact, most copper compounds, are capable of depositing metallic copper on brass. Less easily understood is the action of basic chloride of zinc. On painting tubes with this body and immersing them in warm sea water a distinct coppery stain results after some time. Dezincification may, therefore, be regarded as a normal process, which, so long as it remains superficial, as in the case of the 70-30 brass, may even be protective.

But there is something more to be said about dezincification. If a portion of the coppery film be removed so as to expose the brass, it does not immediately become recoated with copper. A tube which had been dezinced to a depth of less than 1/500 inch was taken, and a small patch about ¼ inch square removed with fine emery paper without appreciably altering the thickness of the tube. On immersing it in sea water kept warm the brass rapidly corroded at this point, until in about a month it was about 1/32 inch deep; copper then gradually encroached on the bare brass and stopped the action. I will refer to this later on.

One of the commonest and perhaps the most serious forms of corrosion is that which leaves the tube scored with more or less continuous and parallel lines, involving the whole or any part of the interior of the tube, often to such an extent that it has a fluted or reed-like appearance; finally, perforation occurs, or the tube splits along one of the lines of corrosion. This type of breakdown is most certainly due to a defect in the manufacture of the tubes. Castings made with a sand core are very liable to have spills, which may extend to a considerable depth in the metal, and which on boring out remain undetected. On drawing into tubes these spills become elongated into what are known as laminations, which may often be traced for a number of feet, in some cases almost through the thickness of the tube. Laminations are comparatively rare on the outside of tubes, since they are more readily detected there, and if present in the casting, it can be turned down until solid metal is reached. On sawing a laminated tube lengthwise the defects appear as shown in the photograph (Fig. 1), and might easily be mistaken for a mere discoloration of the metal. On flattening the tube, however, they open up, showing their true character. Fig. 2 is a photograph of such a flattened tube. These defects may occur as mere surface folds, or may penetrate the wall of the tube to any extent. On opening them up they are seen to be coated with oxide of copper, which, on immersing in sea water, is grad-

ually reduced to the metallic state; and this is probably the initial stage of electrolytic corrosion. Sooner or later the corroded lamination breaks away, exposing fresh lines of yellow brass, and the action is concentrated at these places (see Fig. 7.) The following experiment was carried out in order to test the truth of the above. An Admiralty tube which showed indications of a lamination was immersed in estuary water, which was changed almost every day over a period of more than two years, the containing flask being kept at a temperature of about 110 degrees to 150 degrees F. by means of an electric heater. After a few weeks the tube was completely coppered, and the suspected lamination showed unmistakable signs of opening out. In about six months it had corroded away to a measurable extent, and several other lines or laminations, which had not been suspected, showed themselves. The tube at present is scored with several lines exactly like the initial stage of the type of corrosion in question. Throughout the test it was noticed that bubbles of gas (air or CO_2, or both) adhered tenaciously to the laminations and the sharp edges of the tube, in fact, these places appeared to act as nuclei for the gases to separate from the water, the bubbles actually forming there and not adhering by accidental contact. Under such a test the rate of corrosion was, of course, very slow, and had it been under actual working condenser conditions the total effect would probably have been exceeded in a few weeks. During the corrosion the lamination presented a varying appearance, at one time showing black lines, which would clear up to yellow brass or film over with copper through which minute specks of brass were visible, but at all times the action was greatest at the laminations and edges. It was this tube (Fig. 3) from which the copper patch was removed to which I have previously referred. It is claimed for tubes made from solid castings in which the bore is either drilled or is pierced by hydraulic pressure that laminations do not occur, and that they are not liable to corrosion. The latter part of the claim is possibly too sweeping, but it is reasonable to expect that they will at least be free from reeding. I have no personal experience with these tubes, but I am informed that they go a long way towards solving this particular difficulty.

Another manufacturing defect is responsible for the occurrence of isolated holes in what may otherwise be perfect tubes. In the drawing process small angular fragments of metal are sometimes curled up and nearly detached from the tube, and in subsequent passes are either pressed back into their original position or are completely broken away, leaving a hole. These defects are usually small, but may be relatively deep, and perforation rapidly occurs at such spots when the tube is put into use.

Many new tubes which I have examined have shown minute holes or pores filled with a black material and surrounded with a copper-colored zone; it would almost appear as if corrosion had already started, and I cannot but think that these points will prove active corrosive centers.

Another kind of corrosion which one occasionally comes across, and which is not to be confounded with the last type, also makes itself manifest as isolated holes or patches. In the present case the holes always lie along the bottom of the tube and never on the sides or top. The fact that certain spots only are attacked suggests what is known as local action, since, if the tube is uniform in composition, there is no reason why the corrosion should select any particular spot; but analysis of the metal underlying the defects never revealed any notable difference from the normal alloy sufficient to account for such action. Another possibility was that some foreign body which could form a galvanic couple with the brass had gained access to the tube and set up corrosion. This hypothesis was strengthened by the discovery of iron in the insides of tubes which had broken down after a very short run of work. On opening up the tubes it was found that the whole of the interiors were coated with a

thick, even deposit of ferric hydrate, which at certain spots along the bottom was perceptibly thicker; and under each of these spots small fragments of iron or magnetic oxide were found, each fragment being in the center of a corroded pit. It is important to note that in every case it was the bottom of the tube which was affected. This experience recalls a set of experiments which I carried out some twelve years ago, the results of which seemed at variance to theory.

1. A clean, bright Admiralty tube was fitted up in the well-known form of a Liebig's condenser, the outer jacket, through which salt water circulated, being of glass, so that the action, if any, could be observed. About the middle of the tube a patch of zinc was electro-deposited, the idea being that it would preserve the tube from corrosion. Steam and circulating water were maintained at such a rate that the steam was completely condensed and the circulating water raised about 10 degrees C. After about 60 hours' intermittent run the zinc became almost black and then gradually turned reddish brown, at the same time the tube lost its brightness and showed signs of etching near the zinc. On removing for examination it was found that the zinc had quite disappeared, and its place taken by spongy copper, at the same time the brass in the immediate neighborhood was superficially corroded to a slight extent. It was clear that copper had gone into solution and had been reprecipitated on the zinc, and so far from protecting the brass from corrosion, it had actually initiated it, or apparently so.

2. A similar tube had the point of a steel needle driven into it, and then broken off flush with the tube. This was fitted into the same condenser arrangement and the water and steam conditions kept as before. After about 90 hours' run chlorine was found in the condensed water, and on inspection it was found that the needle point had disappeared, and the hole, which was surrounded by a copper-colored zone, was now large enough to pass the thicker shank of the needle.

3. A helical coil of steel wire was sprung on to another tube and submitted to the same action. After about 120 hours' run it was removed for examination. A distinct print of copper was found marking the position of the coil, which also showed traces of copper where it was in contact with the tube.

4. In a similar test pieces of clean gas carbon were wedged into the water space so as to be in contact with the tube. Within 16 hours the points of contact were surrounded by a coppered area, and in about three days the brass was measurably corroded at these points. The same experiment carried out with common coke gave similar results, but possibly took a longer time.

5. A spot of lead was fluxed on to a tube and filed nearly flush. The whole was thoroughly cleaned and anchored in a pool in the Mersey, near low-water mark, and observations made frequently. A green film gradually formed round the lead, and eventually spread over the whole tube. After a period of about three months the tube was corroded to a measurable extent just at the junction of the lead and brass; but before another observation could be made the tube disappeared.

It is rather remarkable that metals such as iron, zinc and lead, which are electro + to brass, should behave in this manner. The action is probably a secondary effect due to copper being redeposited on the foreign metal. In this connection it is noteworthy that magnetic oxide of iron, or metallic iron coated with this oxide, is sometimes electro + with reference to brass and at other times electro —. The reason for this erratic behavior is yet to be discovered. A form of corrosion of which I have definitely traced the cause is that which attacks the inlet ends for a few inches immediately beyond the tube plate. Here the cold circulating water, on becoming warm, deposits its dissolved gases; the part in the tube plate being kept cool, escapes this particular action, but just beyond this bubbles

of CO_2 and air adhere to the tube, causing round each a ring of corrosion which eventually takes the form of a horseshoe, the open ends of which point in the direction of the water flow. This type of defect is most certainly due to the main condenser being used in harbor for the auxiliary machinery, when the smaller volume of steam does not necessitate the full-power circulation, and the friction of the water is insufficient to sweep away the deposited gases. When an auxiliary condenser is used for port service this defect is obviated. Several rivers and harbors, and canals especially, have an evil reputation in this respect, all being more or less foul with sewage, while those with a good tidal scour are less objectionable. The accelerating effect of sewage, CO_2, and ammonia is well known.

Fig. 4 is a photograph of an opened-out tube which has suffered from this type of corrosion. It will be noticed that although the part in the tube plate is corroded it has escaped the typical horseshoe action. The vessel from which this tube was taken was known to berth in a very dirty harbor. Dezincification is general all over the tube, and in the middle of each horseshoe, but where the corrosion is severe the brass remains yellow. This seems to point to electrolysis between the separated copper and the brass.

There is another type of corrosion which affects the ends of the tubes and which may be found in any part of the condensers, and may occur singly or in groups. The extreme ends of the tubes are eaten away in the ferrules, which also may be involved; and in this case a curious accommodation between tube and ferrule often takes place, the more corroded part of the one fitting into the less affected part of the other, almost as if the action ceased when the pressure between the two had been relieved. This is most probably the case, since in cases of this type I have almost always observed that the tubes had been crushed or distorted by the ferrule. Physical condition as a factor in brass corrosion is fairly well established, and this seems a case which meets this theory. A condenser is generally tubed by several men, each taking a certain section or group of tubes, and it is not unlikely that one worker may have exerted such force as to bruise his tubes. This probably explains why such corroded ends generally occur in groups. So far as my experience goes the defect does not recur on the tubes being replaced. Fig. 5 is a typical case in which both the crushing and the corrosion can be observed.

Some time ago my attention was directed to a case of breakdown in a number of tubes of the non-drip type. These tubes have a longitudinal furrow running along the top, except at the ends where they enter the tube plate. The section is approximately shown in Fig. 6. All the tubes were badly corroded and holed in the position shown in black. As the metal must have been severely strained in the process of grooving and the corrosion was entirely confined to this locality, this would seem a definite case of corrosion due to difference of physical condition. Had the tubes been annealed after finishing it is improbable that this particular defect would have appeared. The great difficulty to a proper understanding of "physical" corrosion is the minute electro-motive force which is called into play, but it must be remembered that the time factor in such cases is a large and dominating one. The behavior of hard and soft brass in an electrolyte is very erratic. The direction of the current so generated varies in an extraordinary manner; sometimes the hard brass is positive and at others negative, apparently due to polarization and the condition of the electrode surfaces. The current generated is never very large, in the cold 2.5 to 5 millivolts. Bubbling air or CO_2 increases the effect, but unless fresh electrolyte is continuously supplied the current falls off very rapidly. Such small currents cannot immediately have an appreciable corrosive effect; but where the conditions are favorable—as in a condenser, where the galvanic elements are in close metallic continuity and the resistance and polarization practically nil—the time factor becomes important. The

E.M.F. from copper-brass is appreciably higher and becomes of significance in the consideration of electrolysis between dezincified surfaces and bare brass.

To my mind, the most interesting, and at the same time the most puzzling, problem is the case where tubes are practically perfect after years of service. I have examined tubes drawn for inspection after eight or ten years' use, which, except for dirt and slight oxidation, have been almost as good as new. Analysis and microscopical examination disclose no reason for this, and whatever the influence may be, it is something quite apart from the metal of the tube, for such tubes respond to artificial corrosion tests just as readily as normal tubes. We are quite in the dark as to the reason for this phenomenon, and any light on the cause would be of immense value both theoretically and practically.

The prevention of condenser-tube waste is almost as vexed a question as the cause. The circumstances governing corrosion are so varied and numerous that what may be found effective in one case might utterly fail in others. Tinning the tubes, provided it is well carried out, seems to add life in many cases, while in others it would almost have appeared to accelerate the breakdown. Zinc protectors in the water ends may have some slight influence over a limited area and time, but as a rule the corrosion of the zinc under such circumstances can be traced to local action rather than to circuit action between the zinc and the surfaces to be protected. The settling of zinc corrosion product in the tubes is, in fact, not unlikely to defeat the object in view. Iron, either as plates or as the material of the water ends, appears to have a favorable influence on the life of the tubes, but how long it remains active is problematic.

These crude methods of opposing a slightly greater E.M.F. have been improved upon in the Cumberland and similar processes with what appears to be success, but, in my opinion, the safer and more satisfactory method would be to replace brass by one of the more resistant alloys which are now available. The initial cost is, of course, higher, but the increased life and relatively higher scrap value of alloys, such as cupro-nickel, renders this drawback more apparent than real. The ease with which brass lends itself to casting, machining and general working no doubt accounts for its use in many situations where other alloys, or even iron or steel, would be of greater advantage as regards corrosion. If designers would only accept the fact that under certain circumstances it is the nature of brass to corrode more or less rapidly in sea water, a great deal of annoyance and expense might be avoided. It is useful to remember that the higher the percentage of zinc in a brass the more easily it is corroded, and that air, CO_2, ammonia and sewage in the water hasten the action, especially in the presence of heat.

The practice of cleaning tubes by acids, alkalies, or some of the secret nostrums which are on the market, is questionable. If the vacuum is good and the corrosion product a thin, firmly adhering scale, I do not think it should be disturbed, since it is more likely to be protective than otherwise. On the other hand, if the tubes are choked to such an extent that the efficiency of the condenser suffers, there appears to be no other remedy than to clean them, but in any case if the deposit can be removed by a high-pressure water jet, or by brushing, this is undoubtedly better than applying chemicals.

Boiling out the steam space of a condenser with alkalies is a practice for which some engineers have an obsession. In turbine sets, where the only oil which can reach the condenser comes from the reciprocating auxiliary machinery, this seems hardly necessary, more especially since the modern tendency is to reduce cylinder lubrication to a minimum, and the oils used are mineral hydro-carbons, on which soda has no action beyond a possible slight emulsifying effect. A slight film of mineral oil can do no harm, and I have even suspected this as the protective influence on those

tubes which resist corrosion, such tubes as I have examined having had a slight greasy coating on the inside.

The steam side of tubes rarely corrodes, but when it does the cause is usually obvious. It is a fact, however, that when a condenser has been laid off, or where air enters with the steam, the condensed water always shows traces of soluble copper for some little time after. This may be the orgin of the trace of copper which is nearly always found in boiler deposits, although it might equally well be derived from the boiler mountings. It is rather a coincidence, however, that old engineers synchronize the first serious boiler corrosion with the introduction of the surface condenser.

In conclusion, I wish briefly to refer to the common idea that leakages from electric-lighting circuits are accountable for condenser-tube corrosion. That such cases have occurred is possibly correct, but in a steamship the free paths open to vagrant currents are so numerous that electrolytic conditions are practically non-existent. Even were such leakage directly through the condenser it would be conducted by the metal rather than through the water. Most of the cases which have been ascribed to stray currents must be attributed to quite other causes.

A precise knowledge of the inner mechanism of brass corrosion will be of great value to engineers and metallurgists alike, but as the present difficulty is not the slow natural decay of the tube in salt water, but the extraordinarily rapid local breakdown, I cannot help thinking that careful and intelligent observation of actual cases will lead to more useful information than laboratory experiments.—" Engineering."

REDUCTION GEAR DEVELOPMENT.

The following clippings from various sources (all written by Mr. John H. MacAlpine) have been compiled for their historical value:

The development of the turbine and later of the geared turbine are far the most outstanding advances in steam engineering in many years. The modern reciprocating engine is fundamentally the invention of James Watt, whose fertile brain and great mechanical skill, exerted over a long life, effected a large number of improvements. Greater steam economy was made possible by the introduction of the compound engine, invented by Jonathan Carter Hornblower, a contemporary of Watt. Further improvement in the same direction was made by Dr. Alex. C. Kirk when he designed the triple-expansion engines of S. S. *Propontis*. The success of the triple-expansion engine would no doubt have dated from the trials of this ship in 1874 had she not been fitted with water-tube boilers (not of Dr. Kirk's design which proved a failure. This success was delayed till the trials of S. S. *Aberdeen* in 1881, and it is one of the writer's proudest memories that he worked on Dr Kirk's staff in the design of these engines. I look back with the deepest gratitude to many years of intercourse and increasing intimacy with this distinguished engineer. The quadruple-expansion engine followed almost at once, but with doubtful advantage. All other improvements, though great in the aggregate, can only be looked on as improvements in detail.

But in 1884 the handwriting appeared on the wall when the Hon. (now Sir) Chas. A. Parsons introduced his reaction steam turbine. From the earliest times it has been the dream of the engineer to produce rotary motion directly by the action of the steam, but the practical solution of the

problem awaited his inventive genius. His reaction turbine soon secured many applications on land, and, after a good many years, very wide adoption in fast steamships. Later, a practicable impulse turbine was developed, and, subsequently, combinations of these two types and compound turbines appeared in great variety.

With one exception the turbine, in the early marine applications, was directly connected to the propeller. This arrangement allowed of very high powers being applied, but even in high-speed ships there had to be an unsatisfactory compromise between the best speed for the turbine and that for the propeller. The former, using a very light fluid, requires a high peripheral speed for the best efficiency; and the latter, working in a dense medium, had to be overspeeded. This always resulted in the adoption of too small a propeller and very large turbines of low peripheral speeds, necessarily inducing great trouble from expansion, stripping of blades, etc. No great advance in steam economy, if indeed any, could be claimed even at full speed; and as the speed was reduced the consumption Gutaf de Laval. It has met with considerable success in sizes up to about the same as for the reciprocating engine. This solution was not at all applicable to other than fast ships.

It was perfectly obvious that if gears could successfully be interposed between the turbine and propeller a solution, extremely satisfactory in all respects, would be secured, and this suggestion was made many times. Indeed, in 1889 a simple geared turbine was introduced on land by Dr. Gutaf de Laval. It has met with considerable success in sizes up to about 200 horsepower; but its long, slender pinion, subject to considerable torsion and not sufficiently protected against cross bending, seems to me to be the defect in the design which presents its application to other than small powers and must be liable to cause trouble even there. It has never been applied to ship propulsion, but in 1897 the Parsons Marine Steam Turbine Company, Ltd., fitted a geared turbine of 10 horsepower on a launch. In the same year the Hon. Chas. A. Parsons applied a gear of the De Laval type, of about 300 horsepower, to drive a dynamo. Though the gears were poorly cut and made considerable noise, the efficiency was found to be over 98 per cent. But in 1901, after, apparently, numerous experiments, we find that he had concluded that gears, except for small sizes, were wholly impracticable.

It was at this stage that the problem was taken up by my late partner, Admiral George W. Melville, ex-Engineer-in-Chief, U. S. N., and myself, resulting in the invention of the floating frame. This is so fully detailed under the heading, "Origin and Trials of the Floating Frame," in the more recent of the following papers, that it is unnecessary to make more than a short reference to it here. Supposing that the teeth could be cut with a near approach to perfection, which seemed by no means impracticable, what appeared to us the greatest difficulty remained untouched: very slight error of alignment of the gear and pinion axes, such as would be sure to arise in operation, would cause imperfect tooth contacts and greatly exaggerated stresses in the metal and at the surfaces of contact. Nevertheless, after much discussion, we made a design of an ordinary gear and pinion of large size, the idea of the floating frame not having arisen. The diameters of the gear and pinion were 70 inches and 14 inches, respectively; indeed, all the dimensions were the same as those of the experimental floating frame gear very fully treated of in the following papers. This we very conservatively estimated would carry a safe load of 2,500 horsepower at 1,500 revolutions per minute of pinion. The late Mr. George Westinghouse proposed to build and test this gear; but while estimates were being sought for building the gear and cutting the teeth there was ample time to critically examine the many important questions involved in the design. The results of this examination were set down in

a document dated February 26, 1906, called "Comparison of Reduction Gear for 2,500 Horsepower with Gear of 300 Horsepower De Laval Turbine." This was sent to Mr. Westinghouse and others, but not published. While making this investigation the excessive delicacy of such a gear to errors of alignment became very prominent, and in this document occurs the first reference to the floating frame—"We have thought very seriously of trying to obviate this difficulty of wear (of the bearings) by carrying this pinion on a very stiff frame so mounted that it would be free to have slight angular motion, both horizontally and vertically, about the center of length of the pinion. The pinion then having three degrees of freedom, would be in proper adjustment however the bearings wore." It was found later that the floating frame would be unstable unless it was deprived of one of these degrees of freedom by means of struts, as is explained in the sequel; but theory also showed that the movement thus restricted was sufficient for the perfect solution of the problem. We now hoped that, having got free from all delicacy of alignment by means of the floating frame, the safe load for this gear at 1,500 revolutions per minute of pinion would be raised to 6,000 horsepower. Experience has shown, as stated above, that this was far underestimated.

Later, Sir Chas. A. Parsons again took up the geared turbine and in 1910 completed S. S. *Vespasian*. Since then he and others have applied it to a large number of ships. But these rigid-gear installations cannot in my opinion compare favorably in space occupied, weight, quietness, efficiency or safety, with the results obtainable with the floating frame.

Origin and Trials of the Floating Frame.—In 1904 the late Admiral George W. Melville and the writer visited Europe to examine, for the late Mr. George Westinghouse, the marine turbine situation. He was also in Europe at this time and we were frequently together. In discussing the question we often advised him that, in our opinion, gearing offered the only hopeful solution which would make the turbine applicable to ships of moderate or slow speed and greatly improve their application to fast ships. But if the gearing was to be of practicable size the pinion became so broad in proportion to its diameter that it was quite evident there would be a great delicacy of alignment, which would limit seriously the power which could be transmitted safely. Hence we made no direct reference to gearing in our report. But later a rigid-gear design was made, and a close study of this made it increasingly certain that the rigid gear left much to be desired. It also became evident that if the pinion could be put under control of the large gear, except as to its rotation on its axis, delicacy of alignment might be overcome. Only long-continued study showed how perfectly this could be accomplished. Hence the floating frame developed, and what I have said in the foregoing shows how completely the solution is, not only regarding alignment, but in diminishing the effects of torsion. The condition required is that moments of the tooth pressures, and therefore also those of the bearings, must be zero about some point I between the helices, and preferably in the center. This cannot be accomplished by any other mode of suspending the bearings or by making the large gear flexible.

The first idea of the floating frame suggested itself towards the end of 1905, but did not fully develop for a considerable time thereafter. We submitted it to Mr. Westinghouse in a drawing dated September 26, 1906, which shows the floating frame, pinion and I-beam almost identically as they were made in the experimental gear; the flexible shaft and loose coupling driving the pinion developed later. Mr. Westinghouse was immediately struck by the advance this opened and asked us to make a complete design of a gear which he could test. This we at once did, but the building of it was delayed by various circumstances, especially by the panic of 1907. After the exceptionally successful trials of 1909, Mr. West-

inghouse wrote in the "Broadening the Field of the Marine Steam Turbine" —a volume privately but widely distributed—page 10: "The certainty with which the floating frame of the Melville and Macalpine reduction gear operates to maintain an evenness of tooth pressure and the limitation of the maximum pressure to 450 pounds per inch of tooth contact with a load of 6,000 horsepower, coupled with a large factor of safety, at once removes this invention from an experimental to a completely commercial apparatus.

"I regard this invention as epoch-making in its importance. It has been my privilege to supply the material things which were needed to transform the creature of Messrs. Melville and Macalpine's imagination into an actual thing of iron and steel. The results achieved by the completed machine have fully justified my faith in its ultimate success."

POWER LOSS IN SCREW PROPELLERS DUE TO ROUGH SURFACE.

Experiments described in a paper by Naval Constructor William McEntee, U. S. N., at the recent annual meeting of the Society of Naval Architects and Marine Engineers, indicate that a great saving of power is effected when ship propellers are made of some non-corrodible metal like bronze and are carefully machined and polished, as compared with the common practice of using propeller castings as they come from the foundry. It is likely that these experiments teach a valuable lesson, also, to the designers and builders of centrifugal pumps. The skin friction of a rapidly-moving impeller in a centrifugal pump must be very great, especially in pumps working at high heads; and it seems all but certain that this skin friction must be greatly increased if the impeller is a rough casting.

Usually screw propellers used on naval vessels are carefully finished by machining the rear surface of the blade to a true helicoidal surface and finishing by hand the front face so that it will be perfectly smooth. The propellers of merchant vessels, however, generally have blades of cast iron or cast steel and are given no finish whatever, except trimming off the rough places with chisel or file. In addition, as is well known, iron and steel propellers often suffer by pitting, so that they present an extremely rough surface to the water.

With the object of determining the relative efficiency of a smooth-finished propeller and a propeller left with the rough surfaces of the original casting, four propellers, 16 inches in diameter, were tested at a uniform speed of 5 knots. Two of the propellers were of bronze, one of cast iron, and one of cast steel. One of the bronze propellers was finished perfectly smooth; the others were left as they came from the mold, with only the ordinary cleaning. The test showed that the propellers with a rough surface had a maximum efficiency of about 63 per cent., whereas the polished propeller had an efficiency of about 72 per cent. Another test was made to determine the effect of a still rougher surface, such as exists on a propeller that becomes badly pitted or on which barnacles and other sea growths accumulate, by varnishing the polished bronze propeller and applying to the sticky varnish a layer of granulated cork, such as is used for cork paint in ship work. The efficiency of the artificially-roughened propeller which had been about 72 per cent. in its polished condition was reduced to 36 per cent.

THE MEASUREMENT OF SHAFT HORSEPOWER.

The use of the steam turbine as a means of propulsion for all classes of steam ships, which bids fair to be considerably extended in the near future as a result of the successful application of gears for the reduction of the most economical speed of the turbine to the most economical speed of the propeller, has rendered it essential that some simple means shall be found whereby the horsepower developed by the turbine can be measured. At the present time there is no direct method similar to the simple and straightforward "indicating" of the reciprocating engine that can be used for determining the horsepower of the turbine, and hence indirect methods must be resorted to. A rough approximate computation can be made, based upon comparison with the known performance of similar machines, or in the case of turbines utilized for driving electric generators, the output of the generator can be measured and, making allowance for the efficiency of the generator, the horsepower developed by the turbine is arrived at.

For marine work, the method adopted consists of a determination of the horsepower being transmitted through the tunnel shaft between the turbine and the propeller, by measuring the angle of twist of a known length of the shaft, the result being known as the "shaft horsepower." It can be readily shown that the angle of twist or angle of torsion varies directly as the power transmitted by the shaft, and inversely as the speed of rotation. Let Fig. 1 represent a length l of a shaft transmitting a horsepower

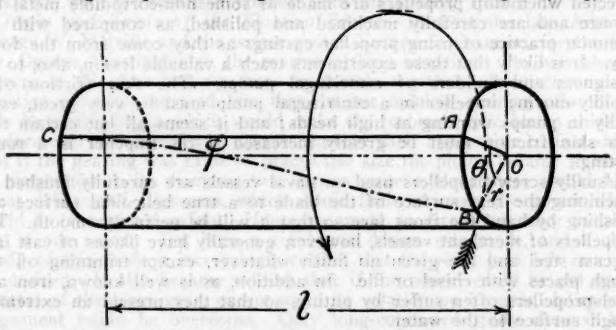


FIG. 1.—DIAGRAM SHOWING EFFECT OF TORSION.

H.P. at a speed of N revolutions per minute in the direction shown. The material of the shaft is subjected to a pure shearing stress as a result of the torsion in the shaft, and the line which was originally in the position CA moves around to CB, CB being a portion of a helix. The radius OA at the same time moves round to OB, so that the angle AOB $= \theta =$ angle of torsion of the length l.

$$\text{But modulus of rigidity } C = \frac{\text{shear stress}}{\text{shear strain}},$$

$$\text{hence, if } f = \text{max. shear stress in the material of shaft } C = \frac{f}{\frac{AB}{l}},$$

$$\text{since shear strain} = \frac{AB}{l}. \quad \text{But } \frac{AB}{OB} = \theta \text{ or } AB = r\theta.$$

$$\therefore C = \frac{f}{\frac{r\theta}{l}} = \frac{f}{\frac{d\theta}{2l}} \text{ or } \theta = \frac{2fl}{dC}. \qquad (i)$$

But it is proved in any text book dealing with the theory of shafts that for a solid shaft of diameter d

$$T = \frac{\pi}{16} d^3 f,$$

where T is the twisting moment producing the stress f. Hence substituting the value of f obtained from this equation, *i.e.*, $f = \frac{16T}{\pi d^3}$,

in equation (i) we get $$\theta = \frac{32Tl}{\pi d^4 C}, \qquad \text{(ii)}$$

But the horsepower transmitted by the shaft at N revolutions per minute sets up a twisting moment T in the shaft such that

$$\text{H.P.} = \frac{2\pi NT}{33{,}000} \text{ or } T = \frac{33{,}000 \text{ H.P.}}{2\pi N}.$$

The unit of twisting moment as obtained from this formula is the pound-foot. This must be expressed in pound-inches, since modulus of rigidity and diameter of shaft are expressed in inch units. Hence we have

$$T = \frac{33{,}000 \times 12 \times \text{H.P.}}{2\pi N} \text{ pound-inches.}$$

Substituting this value of T in equation (ii) we get

$$= \left(\frac{32 \times 33{,}000 \times 12 \times l}{2\pi^2 d^4 C}\right) \frac{\text{H.P.}}{N}. \qquad \text{(iii)}$$

But for any given shaft l, d and C are constants

$$\therefore \theta = \text{constant} \times \frac{\text{H.P.}}{N}, \qquad \text{(iv)}$$

and a similar result would be arrived at if the foregoing investigation had been made for a hollow shaft instead of for a solid shaft. Hence in either case, having determined the constant of the formula, the horsepower can be directly determined from a measurement of the angle of twist of a given length of the shaft.

The instrument used to measure this angle of torsion is called a torsionmeter, and in most types on the market consists essentially of two collars clamped rigidly to the shaft at a given distance L apart, as shown in Fig. 2. The second of these collars is provided with a sleeve and a flange at the end of the sleeve, which is thus brought up close to the first collar, which carries a similar flange, the sleeve overlapping the first collar on which it is supported, usually on roller bearings, to keep it concentric with the shaft. Both the collar and the sleeve are quite rigid, and it is therefore obvious that when the shaft is twisted by the transmission of power, the flange on the sleeve will move relatively to the flange on the first collar, the movement being equal to that between the two parts of the shaft to which the collars are clamped. Actually the relative movement of the flanges is very small, and it is in the means adopted to multiply and measure this relative movement that the various types of instruments differ.

In the Hopkinson-Thring torsionmeter (see Fig. 3), made by Siemens Bros. & Co., Ltd., and designed by Professor Hopkinson, in conjunction with Mr. Thring, the principle adopted consists in the observation of the twist by means of two beams of light projected on to a scale from a fixed and a movable mirror. It is of interest to note that a somewhat similar result was arrived at independently, although at a later date, by Hermann

Frahm, of Hamburg. The beam projected on to the scale by the fixed mirror is taken as the zero point, whilst the beam projected by the movable mirror indicates the amount of torque on the shaft. Both mirrors rotate with the shaft, but even at moderate speeds the reflections appear as continuous lines of light across the scale, Fig. 4, and there is thus no difficulty in taking the readings. The fixed mirror is attached to the flange on the first collar referred to above, and the movable one is attached to the flange carried by the sleeve. The fixed mirror is adjusted so that the beam of light reflected from it is received at the same point on

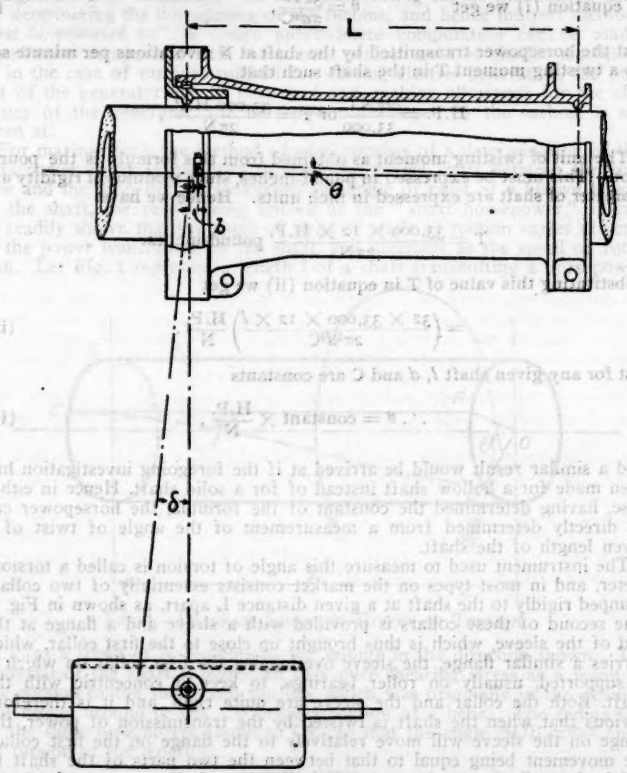


FIG. 2.—DIAGRAM OF TORSIONMETER SHOWING ARRANGEMENT OF FLANGES.

he scale as that from the movable mirror when there is no torque on the shaft. If the position of the apparatus becomes altered relatively to the scale, owing to the warming up of the shaft or from other causes, this is indicated immediately to the observer by an alteration in the position of the zero as reflected by the fixed mirror. Hence the scale zero may be adjusted, if desired, by moving the scale so that its zero coincides with the reflection from the fixed mirror, but this is not necessary to obtain a correct result, since the mean of the two readings will be the same.

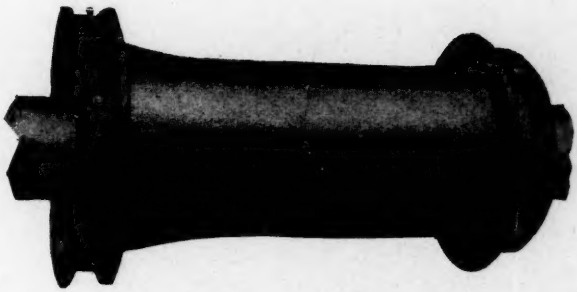


FIG. 3.—HOPKINSON-THRING TORSIONMETER.

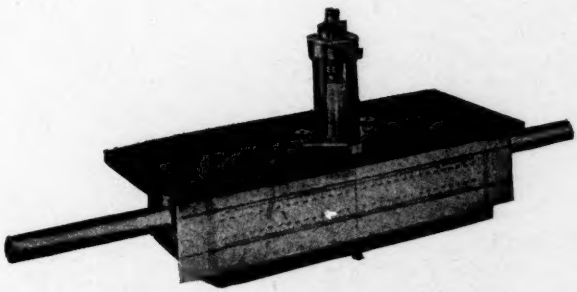


FIG. 4.—INDICATOR FOR HOPKINSON-THRING TORSIONMETER.

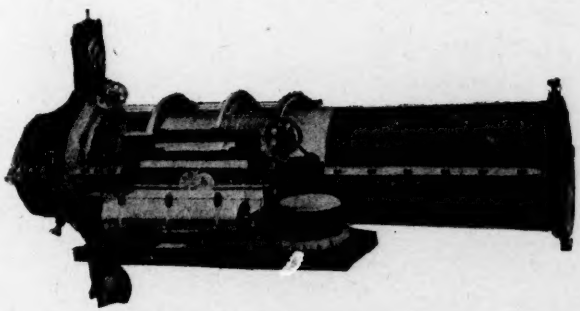


FIG. 5.—DENNY-EDGECOMBE TORSIONMETER WITH INDICATOR AND RECORDER

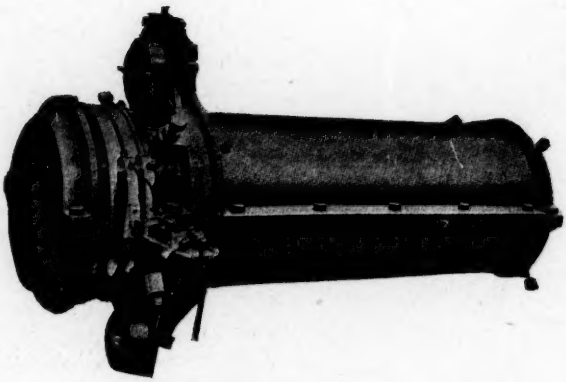


FIG. 6.—DENNY-EDGECOMBE ELECTRO-MECHANICAL TORSIONMETER.

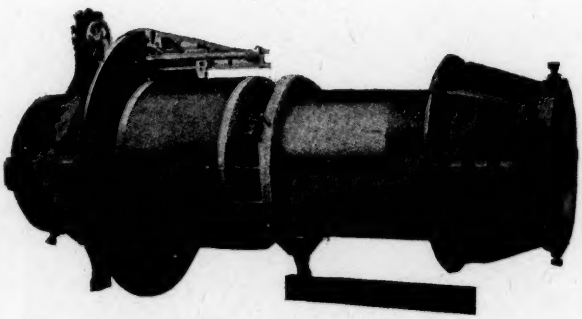


FIG. 9.—DENNY-EDGECOMBE RECORDING TORSIONMETER.

The constant of the instrument, viz.: the factor which when multiplied into the product of the torsionmeter reading and the revolutions per minute gives the horsepower may be determined either by calculation or by experiment. It is determined by calculation from the various quantities given in formula (iii) above, modified if necessary in the case of a hollow shaft, together with a factor giving the relationship between the scale reading and θ the angle of twist of the shaft; but a direct calibration of the shaft with the instrument in position is recommended, this being made before the shaft is put in the ship by applying a known twisting couple.

The angle of twist of the shaft is equal to the angle between the two beams of light. If this angle is small, it is approximately proportional to the scale reading.

We have seen that

$$\theta = \text{constant} \times \frac{\text{H.P.}}{\text{N}}, \quad \text{(iv) above,}$$

i.e., $\text{HP} = \text{constant} \times \text{N}\theta$; and since θ is proportional to the scale reading

$$\text{HP} = \text{constant} \times \text{revs. per min.} \times \text{scale reading.}$$

The value of the constant is best determined for different values of the twisting couple, these being in approximate agreement, and the average value taken for use when the instrument is in service on board the ship. If, however, the torsionmeter is applied to existing shafts, where it would be impossible to make direct readings in order to obtain the constant, this can only be determined by calculation, as shown above.

By a combination of mirrors it is possible to obtain more than one reading of the torque in the shaft during each revolution, but it will be apparent that this instrument is only suited to deal with shafts in which the twisting movement is uniform, as in turbine shafts, as in the event of a reading being made when the torque was very much different from the average value, the shaft horsepower arrived at would be correspondingly inaccurate.

Another form of instrument is the Denny-Edgecombe torsionmeter, made by Denny Brothers, of Dumbarton, the principle of which is the same as that already described, but which differs in the means adopted for measuring the relative displacement of the two flanges. Each of the flanges carries two similar and similarly situated arms, and when the shaft is twisted, the sleeve remaining rigid, there is a transverse displacement of the arms carried by the flange attached to the sleeve, relatively to those carried by the other flange, the displacement corresponding to the twist of the shaft between the two collars. One of the arms has a short rack of involute teeth on its periphery, the other carries an enclosed multiplying gear, the pinion of which can be accurately adjusted to mesh with the rack. This multiplying gear is really a set of toothed levers which have small angular movements upon their centers, but for convenience they are made in the form of wheels, both for accuracy of manufacture and for balance. There are only two spindles, one carrying the pinion which gears with the rack; the other has the final angular movement which it is desired to measure. In the type of torsionmeter represented by Fig. 5, a mechanical contrivance is adopted for measuring the angular movement of this second spindle, there being fixed to it a balanced drum about 4 inches in diameter, as shown. To this drum there is secured by both ends a flexible wire rope, which passes from the drum down the arm round a pulley along the sleeve, round a horizontal pulley on the sleeve and back again to the drum, to which it is securely fastened, the pulley on the sleeve being adjustable in order to tighten the rope. Thus the circular motion of the second spindle is converted into a longitudinal movement

of the wire rope. A light aluminum traveler runs on rollers along a turned part of the sleeve, one pair of its rollers being made to run in a V-groove cut in the sleeve, thus permitting the traveler to move along the sleeve, but causing it to revolve with the sleeve. The traveler has a turned flange and a gripper, by means of which it can be rigidly secured to either part of the wire rope in any desired longitudinal position. It is evident that when the shaft twists, the multiplying gear is moved, and the traveler moves along the sleeve an amount which represents the torsion of the shaft.

There may, however, be a slight longitudinal movement of the whole shaft due to expansion or other causes, and it is therefore necessary to measure the movement of the traveler relatively to the sleeve. For this purpose there is cast upon the sleeve a base flange, which is turned to the same diameter as the traveler flange. This forms the base of measurement and also serves as a means of accurately setting the traveler on the sleeve, the two flanges being brought together for this purpose, so that when the traveler is secured to the wire rope, the difference of position which occurs between the two flanges represent torsion accurately. A suitable indicator is attached to the traveler, which multiplies the motion fourfold, and thus a more accurate measurement of the relative displacement of the two flanges—and therefore of the twist of the shaft—is obtained. The indicator consists essentially of a base plate which is free to move longitudinally on ball bearings in its case, and which carries at the left-hand corner a wheel which runs on the base flange, and which is kept in contact with it by a spring so arranged that the indicator base plate follows any longitudinal motion of the base flange and eliminates everything but torsion. Mounted on the base plate is a small movable carriage which carries at the back a similar wheel running on the traveler flange; this flange and wheel are kept in contact by another spring arranged on a fusee to give constant tension to all positions. On the front of the carriage there is a pair of concentric toothed wheels, the smaller of which gears with a fixed rack on the base plate, the larger gearing with a movable rack running in ball bearings, also on the base plate; the longitudinal movement of this rack being four times that of the carriage and traveler flange. The rack carries a pointer which moves over a horizontal scale fixed to the base plate, and so registers torsion. In order to increase the fineness of the reading a small spindle is geared to the edge of the larger toothed wheel, and this moves a pointer round a fixed circular disc, one revolution of this pointer corresponding to two divisions on the fixed scale.

If it is desired to obtain a permanent diagram of torsion, a recorder may be attached to the front of the indicator, as shown in Fig. 5, the drums of which are driven by means of small double-worm gear from the spindle of the wheel which runs on the base flange, change gear actuated by a small lever being provided so that the drums can be kept revolving in the same direction whether the ship is going "ahead" or "astern." A diagram paper is fixed to the drums upon which pens attached to the fixed scale and to the movable rack of the indicator mark respectively the zero line and the torsion line. In addition to this, a clock is fitted with an electrical arrangement for making half seconds, and another electric pen is provided, which can be connected with a make-and-break fitting on the revolution counter of the engine. A diagram of simultaneous revolutions per minute and torsion is thus obtained, from which the shaft horsepower and any or every instant during the trial can be obtained.

It will be noticed that it is the mean torsion which is given by this instrument and not the torsion at any one point of a revolution taken as the mean. As in the type of instrument previously described, having determined the constant either by direct experiment on the shaft or by calculation, the shaft horsepower is determined from the formula

$$\text{S.H.P.} = \text{Constant} \times \text{revolutions per minute} \times \text{scale reading.}$$

One of the difficulties encountered when using the instruments just described is due to the limited space available in the shaft tunnels of many ships, it being necessary with these types that the indicators and recorders shall be mounted near the torsionmeter, which itself is mounted somewhere on the shaft in the tunnel. In order that the readings may be easily taken in any convenient position, an improved form of indicator for the Denny-Edgecombe torsionmeter has been devised. This indicator is of the electro-mechanical type, the relative motion of the two flanges being first of all multiplied mechanically, and then caused to vary the resistance of a rheostat, an electric current passing through which is utilized as a measure of the angle of torsion of the shaft. The indicator itself is connected to the torsionmeter by flexible leads, and can therefore be placed in any convenient position away from the tunnel. An illustration of the torsionmeter is given in Fig 6, and an outline diagram is shown in Fig. 7.

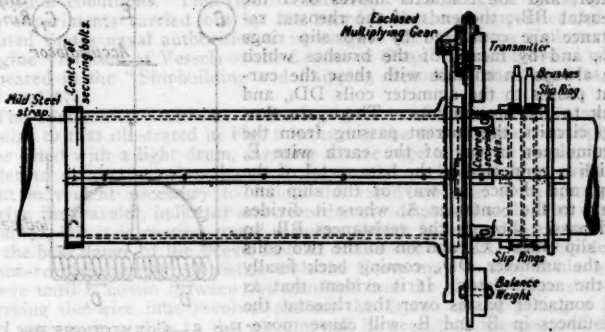


FIG. 7.—OUTLINE DIAGRAM OF ELECTRO-MECHANICAL TORSIONMETER.

The meter has the same fundamental construction up to the second spindle of the multiplying gear as that already described. The second spindle in this case, however, carries a light arm, one end of which acts as a pointer, whilst the other end carries an electrical contacter; and the movement of this arm over a dial fixed to the frame of the multiplying gear is all the work that the gearing has to do. For shafts having a uniform torsion there is no motion of the pointer over the dial for constant revolutions of shaft, but for reciprocating engines the motion is considerable. The dial has an accurately divided scale for about 180 degrees of its circumference, and the pointer can be adjusted to the zero in the middle of this scale. There is a coiled clock spring between the spindle and the dial which eliminates the effect of backlash in the teeth of the gear. When the shaft is being twisted in the shop by means of applied moments the torsion may be read from this scale, but when it is revolving in the ship, it is impossible to read the scale by eye, and electrical transmission is therefore used to show the reading on a stationary indicator situated in any convenient position.

The transmitter consists of a rheostat embedded in the dial, with a contacter passing over it, the contacter being carried on the other end of the pointer arm. The rheostat is a thin strip of resistance ribbon of great

length, arranged in insulated folds and disposed radially around the dial, the folds of the ribbon being extremely close together, thus forming a continuous resistance. Electrical connection is made between the two ends of this resistance ribbon and two slip rings on the body of the torsionmeter, from which the current is collected by two brushes of special design, which ensure uniform transmission of current. The diagram of the electrical connections is shown in Fig. 8, "A" representing the pointer and the contact finger, which is moved round the dial by the multiplying gear. As the shaft twists, A turns on its center, and the contacter moves over the rheostat BB_1; the ends of the rheostat resistance are connected by two slip rings CC_1, and by means of the brushes which are always in contact with these the current passes to the ammeter coils DD_1, and back to the accumulator. There are thus two circuits, the current passing from the accumulator byway of the earth wire E, which is connected to the bare metal of the ship, and thence by way of the ship and shaft to the contacter A, where it divides and passes through the resistances BB_1 to the slip rings CC_1, and on to the two coils of the ammeter DD_1, coming back finally to the accumulator. It is evident that as the contacter passes over the rheostat the resistances in B and B_1 will cause movement of the coil of the differentially-wound ammeter, and the pointer of the indicator will follow the contacter of the rheostat, thus measuring the twisting moment on the shaft. The indicator, *i. e.*, the ammeter has two scales, one being red and the other black, each of the scales corresponding to one-half of the resistance of the transmitter, the scale of which is also half red and half black, so that distinction may be made for "ahead" and "astern" readings. When the meter is set up, the pointer on the transmitter is set to zero, and when the shaft is running the only operation necessary to read torsion is to press the key on the ammeter and take the reading.

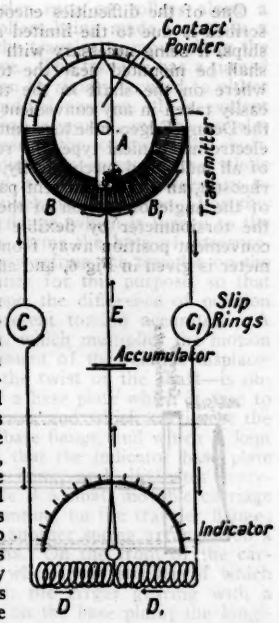


FIG. 8.—CONNECTIONS FOR INDICATOR OF ELECTRO-MECHANICAL TORSIONMETER.

In finding the constant of the apparatus by experiment—which method is to be preferred to that by calculation—the meter is secured to the shaft in the testing shop in the position it will occupy in the ship; the pointer on the transmitter is set to zero when the beams and scale pans, but no other weights, are applied to the shaft. All that is then necessary to do is to apply the loads and read the multiplied angle of twist direct from the transmitter for each increment of load, no electrical transmission being necessary in this case since the shaft is not revolving. Having determined the constant of the instrument, the shaft horsepower is given by the same formula as before, *i. e.*—

$$\text{S.H.P.} = \text{constant} \times \text{revolutions per minute} \times \text{scale reading.}$$

From the foregoing descriptions it will be realized that whilst the twisting moment in a shaft can be easily and accurately determined, provided that it remains fairly constant during each revolution of the shaft,

the apparatus is not suitable for recording the twisting moment in the shaft at every instant during one revolution. It has been felt that with turbines we are working more or less in the dark, and that what is needed is some means whereby the fluctuation of twisting moment in the shaft during one revolution—if any—may be rendered visible just as the fluctuation of pressure on the piston is rendered visible by the indicator. With reciprocating engines the variation of twisting moment is well known, but it is not always correct to assume that because a steam turbine may be supposed to exert a uniform twisting moment, therefore the shaft driving a propeller has no variation of torsion; even with turbine shafting there is often considerable variation under certain propeller conditions.

The Denny-Edgecombe torsionmeter has received a very interesting modification, which enables what may be called "indicator diagrams of twisting moment" to be taken for one revolution of the shaft under any particular conditions. This type of instrument was designed specially for the experiments carried out by the board which was appointed by the United States naval authorities to investigate the "Shafting Failures and Engine Vibration of Vessels of the *Louisiana* Class," a full report of which appeared in the "Shipbuilding and Shipping Record," June 7 and 14, pp. 548 and 574.

The torsionmeter with its indicating gear is shown in Fig. 9, and is similar to that illustrated in Fig. 5 in having the same type of multiplying gear fitted with a light drum, carrying a wire rope along the sleeve, but in order to obtain the advantages of the purely mechanical type, with the light movement necessary to follow the rapid motions of reciprocating shafts, the traveler, indicator and recorder are dispensed with. The wire is retained, but it is carried round a pulley, which is secured to a standard on the base flange of the sleeve, instead of the sleeve itself. By this device a non-revolving drum concentric with the sleeve may be slipped along the sleeve until it comes between the standard and the sleeve. The standard carrying the wire thus revolves around the drum. By attaching a light pen running in a slide to the wire, it will be seen that the pen will trace a line on the drum combined of two separate motions. One is circumferential and is due to the revolution of the shaft, and if there is no torsion, or if there be a constant torsion, a straight line will be marked round the drum. If, however, the torsion be varying, the pen will traverse its slide in a longitudinal direction, so that, supposing the shaft to be twisted but not revolved, a straight line will be marked along the drum parallel with the axis of the shaft. When transmitting power, the shaft is both revolving and being twisted, and the pen then traces out a sinuous line combining the two motions, and thus records the line of variable tension.

It will be seen, however, that the "pen," as it has hitherto been called, is attached to a revolving part of the apparatus, and is therefore inaccessible when the machinery is running; also that it moves over the drum at a speed which may be very high on large-diameter shafting. The method adopted for dealing with this difficulty is to cover the drum with carbon paper, or other material, which will mark a white paper under the pressure of a running pen. Over this carbon paper, on the drum, is secured a sheet of tracing paper. The pen point is a roller with a rounded edge carried on a spring support, provision being made whereby this roller may be pressed upon the drum for one or more revolutions as may be desired, and thus a diagram of the torsional movements is obtained on the back of the tracing paper, but which can be seen through it. There is a second fixed pen in addition to the torsion pen, which gives a base line from which the correct zero line for no torsion can be measured.

It is necessary to know the position of the diagram relatively to the engine, and this can be obtained after the torsionmeter is secured to the shaft by noting the circumferential position of the pen upon the paper

when the engine is at some known position, say with the high-pressure piston at the top of its stroke; this point being noted, it can be transferred to the drum, and thence marked on all subsequent diagrams. It may be found convenient to make the pen position coincide with the joint of the paper for a definite position of the engine, all other positions of the cycle being thus fixed.

The drum is fitted with rollers which run on the base flange, and is kept in position by a spring catch, which adjusts itself to any longitudinal movement of the shaft. The drum therefore follows the shaft, and any error which might be due to longitudinal motion of the shaft is eliminated from the measurement of torsion. The rollers fitted to the drum run on the turned part of the sleeve, so that when the shaft is revolving, the drum being stationary, the friction is reduced to a minimum. The drum is kept from revolving by a casting which slides on a fixed catch bar attached to the ship, there being two lever catches on this bar which hold the drum either against the flange for obtaining the diagram or away from the revolving standard altogether so that the diagram may be removed, and fresh paper put on for the next diagram. The meter is so arranged that diagrams may be obtained either when going ahead or astern.

Although these diagrams are not taken to any great extent by engine builders, it is obvious that much interesting and useful data could be obtained with regard to shaft vibration, effect of propeller, etc., if a series of such diagrams was taken from different classes of vessels and carefully analyzed; and in any case both the maximum and minimum, and also the mean twisting moment can be obtained from the diagrams and from the mean value, the shaft horsepower can be obtained in the usual way.

In conclusion, the author wishes to express his indebtedness to Messrs. Siemens Bros. & Co., Ltd., of Woolwich, and to Messrs. Denny-Edgecombe, of Dumbarton, for their kindness in placing the particulars and illustrations of the instruments made by them at his disposal.—"Shipbuilding and Shipping Record."

DEVELOPMENT AND PROGRESS IN AVIATION ENGINES.*

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The art of aviation and the industry of building airplanes, engines and accessories, to meet the demands of the art, are now going forward with wonderful strides. Commercial usage as well as war usage begins to attract the attention of the aeronautic engineer.

It is interesting to refer back a few years to the "Journal of the Franklin Institute," dated October, 1910, and learn to appreciate the rapidity of change in the art. Reference is made therein to a "useful load of 450 pounds, including three passengers." This weight is the equivalent of about 75 gallons of gasoline, which is less than the amount required for four hours' flight of a two-passenger airplane of recent design and construction. Nearly twice this volume of gasoline is now carried, with two passengers and 150 pounds additional useful load.

Reference is made to airplane flying altitude: "Prominent aviators who undertake a cross-country flight select their course with care and maintain a very high altitude" "Lambert, who ordinarily does not seek high altitudes, in soaring over Paris, was obliged to ascend to a height

* Presented at a meeting of the Mechanical and Engineering Section, held Thursday, March 1, 1917.

of more than 1,600 feet." In the war zone danger is now imminent below 16,000 feet altitude. Much flying is necessary at greater altitudes. Peaceful cross-country flying is done at what are now called safe altitudes of from 3,000 to 7,000 feet. From such altitudes it is probable that a suitable landing area could be reached in case of need. With a "dead" engine the angle of glide is about six to one. This means that a six-mile radius circle is available for a choice of landing from an altitude of one mile.

Duration of flight is referred to as follows: "Present-day flights are no longer measured in minutes." "The hour has been exceeded by many pilots." "The aeroplane will enter the field of touring." All this and more is now true after a period of six years.

Aviation engine weight per horsepower delivered has not changed much; but engines are more reliable, more efficient and stronger, and heavier where experience has taught the need. The term "weight per horsepower" is a loose one. There are many parts that may or may not be considered as a part of the engine of the power plants to be included in the weight. Some twenty-eight or thirty items have been found in this doubtful class. The weight of gasoline and oil consumed per hour is figured by some as an element of engine weight. It does influence the design of the airplane as a whole. It must be known. For example, the rotary, air-cooled engine, although very light, about two pounds per horsepower, is outclassed by heavier engines if the airplane must fly more than three hours. The rotary engine consumes too much gasoline and oil per horsepower-hour for long flight.

Broadly speaking, it may be said that the weight may safely vary from two to five pounds per horsepower. This will include the weight of magneto, carburetor and wiring. Other elements of the power plant, such as water, water piping, oil in the engine and the propeller, are not included.

An engine which ranks very high for speed, climbing and endurance weighs 2.4 pounds per horsepower. This engine is high strung and delicate, but very effective. Many types of engines are in practical use. There is an ever-changing idea as to the proper function for each type, and a constant shifting of opinion of one engine to another of any given type, as to the superiority. Any improvement of design, material or workmanship may result in temporary superiority. Competition in this matter is keen. All sorts of daring expedients are tried under war pressure. The result is rapid change and rapid improvement.

The future one may not predict. The probable status six months hence is only speculation. It is certain that greater horsepowers are appearing for all types and sizes of airplanes. The result is increased speed, climb and load-carrying capacity. There seems to be no limit to this increase except that imposed by lack of experience. This lack is being made good very rapidly just now by war conditions.

About a year ago it was thought that no more than 90 horsepower could be obtained from a radial rotary, air-cooled engine. One-third more has been obtained, and nearly double is now planned.

Similar improvement is being brought about, by experience, in other engines. An eight-cylinder, V-type engine now develops consistently 210 horsepower. Six months ago it did not function as consistently at 180 horsepower. There has been no radical change of design or size.

Experience has been the teacher. No internal-combustion engine of new and extreme design can be brought to a condition of practical excellence in less than a year, and often two years. Yet partially developed engines must be purchased and used or there will never be complete development.

The next, and very important, step is the development of organizations that can really manufacture an aviation engine or airplane, rather than build a few models, or a dozen or two nearly alike. This is a task that

will easily occupy another year of consistent endeavor. At least one aviation manufacturing organization finds itself in this state of development. The building nucleus of men must be expanded into a manufacturing force. The product must be adapted to practical methods of manufacture, to existing tools, and standards, so far as possible. Otherwise no quantity production may be expected.

This has been proved many times in other industries. The aircraft industry is not different. In fact, it is astonishingly similar to the motor-car industry as found during its beginning and early growth.

Perfection would be approached by having each plant producing aviation material, as engines or airplanes, so organized and equipped as to manufacture one type only and do that well. This condition will be brought about by a consistent demand or in response to a pressing need. A proper foundation for such a development is now being laid by the same engineering body that assisted in the amazing growth of the motor-car industry—The Society of Automotive Engineers (recently Automobile, but expanded to include closely-allied industries, of which aviation is one). Its standards and engineering practices will creep into the new industry as they did into the older one.

It is difficult to discuss aviation without mentioning the Wright Brothers. They developed much knowledge of flying without an engine by gliding experiments, and then sought an engine.

At the outset of the automobile industry the power estimated as necessary was too small. Practical tests indicated that three horsepower would suffice, with a generous factor of safety. History repeated itself. Orville Wright recently stated that an eight horsepower engine was figured as necessary. Such an engine, not to exceed 200 pounds in weight, was sought in the open market. This occurred in 1903, and none was offered that was worth considering.

The Wrights then built a four-cylinder 4x4-inch engine that did weigh less than 200 pounds. This engine developed 12 horsepower and actually flew an airplane. This engine was followed soon by a much better one that developed 30 to 35 horsepower. It was simple in the extreme degree. A jet nozzle in the intake pipe served to atomize the gasoline; the ignition was a low-tension system; the water jackets were of sheet aluminum, the crank shaft very small in diameter, as were all other moving parts. Later engines of the same design have produced more power, are very efficient, and still being used to a limited extent.

The four-cylinder vertical engine was the first aviation engine to do real service. The airplane flown with so small an amount of power was large in wing area for the load carried, slow in speed and sluggish in movement, judged by present standards. Such an airplane resembles the butterfly rather than the swallow. It is not like the successful fighting machine, yet will surely find place for peace usages.

Closely following this first type of engine came six-cylinder vertical and eight-cylinder V engines of 65 horsepower and 75 horsepower. All of these were too light and frail for hard service. Minutes of flight were possible, but not hours, as now required. High-pressure lubrication to vital parts had not been developed. Valves failed because of inadequate cooling and material.

In August, 1908, the Wright Brothers made their first flight in France. Following this date the French engineers developed airplanes and engines of various types that embodied the fundamentals demonstrated by the Wrights, so far as the airplane was concerned. The scientist contributed much to the art by working out the theory of the art of flying. A long step forward was taken in the succeeding years.

Air-cooled engines of the radial rotating form were developed, then air-cooled and water-cooled vertical and fixed cylinders. These successive

steps again repeated the steps taken by those who had developed the automobile and its engine; all were too frail and of small power. It seems safe to state that nothing as large as 100 horsepower was reached or thought necessary. The Gnome radial rotating engine became most successful, and, as experience improved the skill of the aviator, the power of the engine grew gradually from 60 horsepower up to 90 horsepower. Similarly the speed of the airplane increased up to 60 or 70 miles per hour, with slow speed or landing speed in the neighborhood of 40 miles per hour.

After a few years of this normal development, with reasonable progress relatively slow because of the enormous cost attending aircraft experiment, came the present European war. Then began a feverish and rapid development, during which rash moves were made and long chances of an engineering character were taken. How much the demands of war have advanced the art of aviation can only be guessed, but surely as much as a generation

As soon as the absolute necessity of aircraft as a part of a modern army was demonstrated the production of them was forced to the limit.

Extreme types are built and flown. Planes of small area, equipped with engines of large horsepower, which must be landed at 80 miles per hour, must be used. This type is the fighting craft that seeks altitudes above the hostile projectile danger zone (about 18,000 feet), and higher yet in order to be able to swoop down on the slower plane or upon some less skillful aviator. One hundred horsepower is no longer enough for this light and agile airplane. An engine, eight-cylinder, V type, of 150 horsepower, is most popular now, but with a tendency toward, and small actual use of, 200 horsepower, yielding a speed of 140 miles per hour and a climb of more than 1,000 feet per minute.

The only type of airplane used for war purposes, satisfied by an engine of 100 horsepower or less, is the school machine, which carries teacher and student, tandem or side by side, in the body. Thousands of these are needed and being used in Europe. It is pleasing to know that one engine built in the United States is giving excellent satisfaction in England for this pu pose.

For large warcraft, battleplanes so-called, more powerful engines are used. So far as known, this demand is met by engines of 250 to 350 horsepower. These aircraft carry bombs or explosives up to 1,000 pounds in weight, and small defensive guns. They are not agile nor fast, and are defended by fast machines, when possible to do so. They are the battleships of the air, defended by the torpedo boat.

To produce this power 12 and 18 cylinders are used. The latter is of the V type, with three rows of cylinders in place of two. It is new and said to be very successful. The power is transmitted to the propeller through gears, the propeller rotating about three-fifths engine speed, at about 1,000 to 1,200 revolutions per minute. Wherever possible, gearing is avoided, because of the great weight added and increased mechanical difficulties. The difficulties surrounding the propeller problem also deter. The combination of much power and high speed tends to wreck propellers. There are many elements entering into this phase of the use of large powers not yet fully understood.

An obvious means to equip an airplane with more power is to use more than one engine unit. This is done and will be done, but there is a penalty attached to the plan. The ideal disposition of weight in an airplane is to have all concentrated at the center of gravity of the entire structure. A single engine unit approaches this condition as nearly as may be. Two units mounted out in the wings, away from the center of gravity, increase the inertia, decrease the agility and maneuvering quality very much. An

airplane so equipped cannot quickly turn and twist to avoid the attack of a fast machine.

There will come a time when all engineering attention is not drawn to airplanes for war purposes. Then a type of plane will be developed that will not have speed, agility and climbing power as its principal attributes. Much less power will fill this need, and a much more normal aircraft will result. Then the multiple-unit plane will have a chance to develop.

Our present aircraft development is what the racing car has been to the automobile industry—a wonderful developer for normal product and usage. In time real manufacture will begin, prices will drop and popularity increase. Engines to meet such a purpose will develop rapidly. They will, as now, be highly refined as to their essential functions, but of ordinary construction as to the non-vital functions. At present the extreme of design, workmanship and material is the aim for all parts. The art is too young to permit any other procedure.

ENGINE DESIGN.

The desired engine must be light, but it must be reliable. It must be capable of continuous operation at very nearly full capacity for long periods. A 20,000-foot climb requires that everything shall be perfect.

The periods of partial use and rest are short, as for a glide. The period of level flying is from 75 per cent. full power to full power, at the desire of the operator. Pursuit or flight means full power, patrol or observation duty means less; but at no time does the usage approach that of the motor-car engine. The demand is even greater than that on the marine engine, which carries a very constant load; but the marine engine may be made heavy—the aviation engine must be light.

Experience has taught that lightness must be abandoned in favor of stiffness and strength for all vital parts, such as crank-shaft, crank-base, main bearings, crank-pin and piston-pin bearings, and that all possible lightness must be gained by hollow construction in combination with materials most carefully chosen. Coupled with these features must be found most thorough oiling at high pressure, with oil led in a positive manner to all vital points. Much experimenting must be done, even after careful design, to arrive at a proper distribution of oil. Clearances, oil apertures and oil viscosity (hot) must be balanced.

Poppet valves must close on seats that are thoroughly cooled. Exhaust valves run cherry-red hot at best. Proportions and detail of design must avoid warping and breakage.

Ignition is desired in complete duplicate for each cylinder. Magnetos must be coupled and driven in such a manner as to minimize vibration. Magneto difficulties have been of mechanical failure more than electrical, and the direct result of excessive strains due to rigid driving couplings. Wiring must be carefully conducted and fastened to prevent failure from chafing, following excessive vibration.

Spark plugs have been the cause of more trouble and uncertainty than any other single item of aviation engine detail. The prime difficulty has been to keep the plug cool. A cool location must be provided in the design. The plug must not extend so far into the explosion chamber as to be seriously flame-swept, nor must it be deep in a hole or recess with delayed and irregular ignition. The plug itself must be gas-tight to prevent "blow by" and resulting overheat. The plug that satisfies the automobile engine will fail quickly under aviation service. This defect is more serious than appreciated, and must be guarded against. It is not easy to locate, as often assumed. Frequent change of plugs is now resorted to in the interest of reliability. Close attention to the foregoing and simple items will surely increase spark-plug life and engine reliability

There is no more mystery about an aviation engine than about any other, when the real truth is known.

One hundred and fifteen pounds per square inch mean effective pressure is very hard to handle in connection with a light bit of construction. This pressure is needed to yield the horsepower per unit volume.

The perfect balance of moving parts is necessary. The airplane structure is very sensitive to vibration, and the occupants are seriously disturbed by an engine not functioning smoothly.

Perfect distribution of gas to give uniform explosions is a requisite. The delicate engine structure fails rapidly under the influence of irregular impulses. Carburetor parts, thin aluminum castings, wiring, gasoline tanks and piping, magneto, radiator and propellers all fail to a serious extent in the presence of vibration.

Simplicity of design, with the smallest possible number of parts, must be the aim. Each additional screw, joint, part or accessory increases chance of trouble in the air. The failure of such detail may cause a forced landing accident—as surely as the failure of a vital part.

The fire risk may be minimized by such design of engine compartment as will promote the movement of a large volume of air in and out of the engine compartment—in such a way as to carry gasoline or gas clear of the airplane body and operator's cockpit. Without an accumulation of gasoline the danger is small.

ENGINE WORKMANSHIP.

The keynote of this feature is that quality must come first, cost second or last. Time must be taken to arrive at the desired accuracy or finish. Time and money must be spent to balance reciprocating parts. A limit of 1 per cent. difference of weight for all pistons and rods in an engine is required in some cases. Exact balance of all pairs is required.

Piston rings must be tight as the result of perfect fit and bearing. Many rings must be discarded to get perfect sets. Rings good enough for motor-car engines will not do the work in aviation engines.

The engine must not throw oil from any joint, valve stem, gear-box or pipe. Oil so thrown may easily cloud the goggles of the pilot. Only the closest of fits will achieve such oil-tightness.

Castings, whether aluminum or iron, must be true and of even wall thickness and of extreme lightness. Many rejections must be made to get the uniformity needed. Such rejections should be figured by those accepting the engine casting orders. The best of pattern equipment and forging dies must be furnished for a comparatively small number of pieces. This adds materially to the cost of a part, but is necessary to attain workmanship of aviation engine quality.

Machine tools, fixtures and methods must be such as to produce absolute accuracy, regardless of cost. "Quality first" should be the watchword in aviation, not "Safety first," because high quality means safety.

It is very hard to create this atmosphere among business men, superintendents, foremen and workmen, who have always been obliged to keep in mind the cost and selling price of a product to meet competition. There is no competition in airplane engines yet. The art is too new. Performance comes first. Only the best in design, material and workmanship will suffice.

It is possible that too much time and money are being spent; but at this time experience has not been great enough to be certain of this. Continued use over a long period of time will determine. During the youth of the industry is not the time to take a chance.

Quality and safety come first.

ENGINE MATERIALS.

Excellent material is required for a motor-car engine, which is rarely worked to anywhere near its limit of power. Take for an example a 40 horsepower engine. It runs quietly about over roads nearly level, with liberal lubricant for the work being done and consuming hardly 25 per cent. (10 horsepower) of its rated power.

Under comparable conditions an airplane engine uses 75 per cent. of its power. Lubrication must be perfect, not nearly so.

Further: a motor-car engine of 672 cubic inches capacity is given a normal rating of 65 horsepower. An aviation engine of the same volume is rated at 154 horsepower. As a motor-car engine not over 25 per cent., or 16 horsepower, would be ordinarily used, but as an aviation engine not less than 75 per cent., or 114 horsepower, would be habitually used, a ratio of about 7 to 1.

In addition, the aviation engine is built as light as possible, say about one-half the weight for a given piston displacement. Roughly speaking, this represents a punishment of material within the aviation engine 14 times as severe as that in the motor-car engine.

The present life of an aviation engine indicates severe punishment. Up to the time when a complete overhaul is necessary this life is about 50 hours, at, say, 100 miles per hour, or a total of 5,000 miles. A motor-car engine runs about 25,000 miles up to the time when an overhaul repair is necessary. The speed cannot average over 25 miles per hour. This means 1,000 hours' operation, or 20 times that of the aviation engine.

A severe test of endurance for the motor-car engine under shop or laboratory conditions, and where destruction of the engine is invited and closely approached, is under a development of power of about 0.15 horsepower per cubic inch displacement. It is considered a fine performance for a stock motor-car engine to last 300 hours under this load. The aviation engine must do not less than this at any time and with 100 per cent. score. Otherwise the engine is not a successful aviation engine.

These are rough comparisons, but they show the relative severity of work. To meet these severe conditions the very best of steel for each engine part must be used. "Best" means that money must not hinder, but high-priced steel is not sure to be best. The highest of technical skill must be utilized: first, to select the steel from chemical and physical soundness standpoints; second, to treat it in such a manner as to give the best possible endurance and the highest elastic limit consistent with the possibility of machining. Here, again, money must not be a consideration. Slow methods may be necessary to work the material.

The crank-shaft of a motor-car engine is usually so large and heavy as to permit the use of steel having an elastic limit of about 70,000 pounds per square inch and with a physical structure not very highly refined by heat treatment. Experience has shown this to be good practice.

The crank-shaft of an airplane engine may be more in diameter, but hollow to such an extent as to demand the highest elastic limit that can be machined, say from 120,000 to 140,000 pounds per square inch, with a reduction of area, denoting the highest refinement of grain, of 50 per cent. or more.

The lighter parts of a motor-car engine, such as valve rocker arms, are given a toughening treatment for a fair forging steel, say 60,000 to 80,000 pounds per square inch elastic limit. The rocker arms of an airplane engine are so light and delicate as to require a fine alloy steel most carefully treated to give resistance to vibration and alternate stress. The elastic limit should also be in the neighborhood of 130,000 pounds per square inch. This, coupled with hardness at certain points, brings additional care and expense.

The highest intelligence and widest experience are needed to arrive at the proper combination of hardness and toughness and endurance. Experimental work to the point of actual failure is necessary to arrive at the extreme limit of endurance. In the aviation engine the extreme is sought.

A decade hence there will exist an accumulation of empirical knowledge. Extremes will then be less common or else the extreme of today will be the commonplace of the future. Experience will repeat in aviation as in many other industries and arts.

THE 260 H.P. MERCÉDÈS.

The engine forming the subject of the present notes is the latest type 260 H.P. Mercédès, and the accompanying drawings and details have been collected from an investigation of the twin engines taken from a captured three-seater "Gotha" biplane of the pusher type.

The engine is naturally of considerable size, and in connection with these larger types of enemy aircraft engine, the 260 H.P. Mercédès is typical of German ideas at the moment. It may be regarded as practically a magnified edition of the smaller ones, and much of the detail design is similar. One of the Mercédès engines was obtained little damaged, with the result that after certain repairs had been carried out it was put through a bench test, when particulars as to B.H.P., consumption, etc., were taken.

In many respects and in general constructural detail these engines resemble the 160 H.P. Mercédès, and follow the usual German practice for aircraft engines of this type. Six separate vertical cylinders, water cooled, with a massive six-throw crankshaft running on plain bearings, are the outstanding characteristics, the design throughout aiming at strength and reliability, combined with ease of manufacture, rather than extremely low weight per B.H.P. Although, of course, of considerable size, the whole engine is of very homogeneous and clean design throughout, as will be seen from the accompanying illustrations. The complete engine, including the propeller boss, measures 6 feet 5½ inches overall, and from the bottom of the sump to the top of the overhead camshaft casing measures approximately 3 feet 10 inches. The bore of the cylinders is 160 mm. and the stroke 180 mm. Four valves—*i. e.*, two inlet and two exhaust—are arranged in the head of each cylinder, and these are operated by an overhead camshaft running in a separate housing, supported on brackets screwed into the cylinder heads; the valves are interchangeable.

The half compression device is operated by sliding the camshaft in the usual manner, and the arrangement is similar to that adopted on the 160 H.P. Mercédès; the actuating gear is fitted to the rear end of the shaft casing.

The induction pipe takes its air supply through the base chamber, to the rear end of which it is attached, and a 4-inch diameter passage is cast in the base for this purpose. A simple type of two-jet carburetor is employed.

The main bearing caps are cast integral with the bottom half of the crank case, and the long bolts that secure them pass through the top half of the crank case, and are used for securing the cylinders by means of triangular clamps.

Forced lubrication is, of course, employed, the bearings and journals being supplied through the drilled crankshaft and suitable oilways. A four-throw eccentric-driven plunger pump of similar but improved design to that used on the 160 H.P. Mercédès type is fitted. An "auxiliary" sump in the front end of the crank case is embodied, and small supplementary pump plungers, which work in conjunction with

the main oil pump, feed fresh oil into the system from the service oil tank. Full particulars of the lubrication system and the somewhat complicated oil pump are given, with other details of the engine, in the pages that follow.

Two Bosch Z.H.6 magnetos are fitted, and these are driven off the vertical camshaft driving spindle. The spark plugs are fitted, of course, on the induction side of the cylinders.

The water-pump driving spindle is lubricated whilst in flight by means of a ratchet-driven grease gun or pump, worked by a cable and lever from the pilot's seat. An electrical tachometer is driven at engine speed from the rear end of the camshaft through a flexible shaft.

At the opposite end of the engine to the half compression actuating gear a small direct-driven plunger pump is fitted for providing air pressure to the fuel tank, while just beneath, and attached to one of the pipes, a trap is fixed to prevent oil, which returns to the sump via the oil-pump casing, getting into the air-pressure supply. A spindle beneath, and on the same center line as the vertical camshaft driving spindle, drives the water pump, which is arranged with a plain dog coupling, so that it may be easily withdrawn should this become necessary for any purpose.

A feature in the construction of these engines is the general use of soft mild steel for such parts as the lubrication pipes, etc. This feature may lead to the fallacious conclusion that the enemy has adopted mild steel owing to shortage of copper, whereas this practice has been standard with the Mercédès Co. for many years—certainly since long before the war.

Such is the general layout of the engine; the following are the more intimate details.

CYLINDERS.

The general construction of the built-up cylinders, composed entirely of steel machined all over, and sheet steel pressed to the formation of the water jackets, presents an interesting example of expert acetylene welding. The building-up methods applied to the cylinders and their general construction are clearly shown in Fig. 1. The cylinder barrels are screwed into the cylinder head, the pitch of the thread being 1.75 mm. They are machined from forgings or billets, and the cylinder walls taper in section from 3.5 mm. at the top to 6 mm. at the holding-down or base flange. Six stiffening ribs are arranged, the distance between the ribs increasing towards the base of the cylinder. The barrels extend 35 mm. below the base flanges, and are of 3 mm. thickness for the depth of 12 mm., forming the register, the extension being then reduced to 2.75 mm., presumably as register clearance and to reduce the amount of limit machining.

Steel forgings are employed for the cylinder heads, and into these are built the four valve pockets, and also the inlet and exhaust ports. The seatings for the valves are machined in the cylinder heads, and the thickness of the crown of the heads is 11 mm. The valve pockets are also machined from steel forgings, and are acetylene-welded into the cylinder heads. Steel valve-stem guides are welded into the valve pockets, and phosphor bronze bushes are pressed into the guides. It will be noticed that the exhaust-valve guide is arranged with greater water space than the inlet.

The water jackets are built up in four sections from sheet-steel pressings 1.25 mm. in thickness, the lower section of the jacket being of barrel formation and welded to the flange joint on the cylinder walls. The top sections are in halves and encircle the valve pockets, the joints being

welded vertically on the center line of the valve ports. A sheet-steel disc is welded on to the flanged top sections to form the top of the jacket, and the water-circulation pipe connections are welded into the top and bottom of the jackets on the exhaust side. The spark-plug bosses are fitted and welded into the cylinder barrels on the induction side just below the inlet valves.

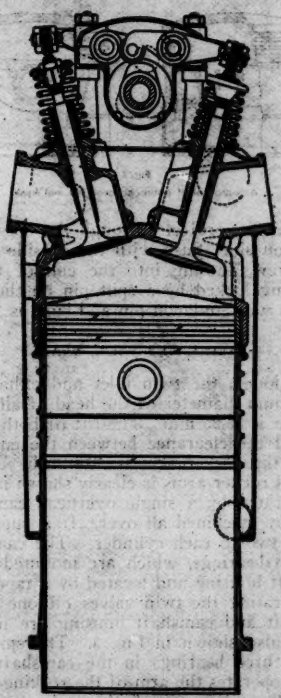


Fig. 1.
Cross section through cylinder and valve gear.

As in the 160 H.P. Mercédès engines, the exhaust valve guides are water-cooled by recesses formed in each of the exhaust valve pockets, so that the water is led right up to the valve-stem guides. Considering the size of the cylinders, they are remarkably light in weight. The complete cylinder, with valves and valve springs, etc., and including the two brackets screwed into the cylinder head to carry the camshaft, weighs 34.25 pounds.

PISTONS.

These are constructed in two parts—namely, the head which carries the gudgeon pin and the skirt of the piston into which the head is screwed. A steel forging machined all over forms the head, the thickness of the crown being 8.5 mm.; it is slightly domed, as shown in Fig. 2. The skirt of the piston is screwed on to the head and acetylene welded at the joint, cast

iron being used for this portion. Three compression rings are fitted in grooves in the cast-iron skirt above the gudgeon pin, while a scraper ring is fitted at the base of the piston. The width of the rings is 5 mm., and all the rings are split with a simple diagonal gap cup at 45 degrees; no locating pegs are fitted to maintain their positions. The gudgeon pin,

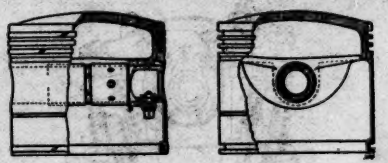


Fig. 2.
Arrangement of piston, gudgeon pin, and bush.

which is 37 mm. diameter, projects through the bosses in the piston head and fits into the piston skirt flush with the outside diameter. It is fixed by an 8 mm. set screw, passing into the end of the gudgeon pin and through the boss, being locked by a split pin on the inside. The weight of the complete piston with gudgeon pin and bush is 11 pounds.

VALVES AND VALVE GEAR.

As previously mentioned, the twin inlet and exhaust valves are interchangeable, the maximum diameter of the head of all valves being 60 mm., while the port diameter is 55.25 mm. The lift of both the inlet and exhaust valves is 10 mm., and the clearance between the end of valve stem and adjustment screw of the rocker arm is .018 inch. The general arrangement of the valves and rocker arms is clearly shown in Fig. 3.

The valves are operated by a single overhead camshaft, enclosed in a housing of special alloy machined all over. It is supported on T brackets on the cylinder head, two to each cylinder. The camshaft runs on seven phosphor bronze plain bearings, which are mounted in aluminum bushes fitted into the camshaft housing and located by a tapered grub screw.

The method of operating the twin valves off one cam and the general design of the camshaft and camshaft housing are interesting; details of this arrangement are also shown in Fig. 3. The spindle of each rocking lever is mounted on three bearings in the camshaft housing, and is so designed that the cam operates the arm of the rocking lever inside the camshaft housing, whilst the other arm of the rocker operates the valve outside the casing, this arm working between two outer bearings of the rocker spindle situated between the separated compartments of the camshaft casing. On the end of the outer rocker arm a double-branch tappet arm is fixed which operates the twin valves. (See Figs. 4 and 5.) Although these branch arms should not move upon the valve rocker, they are not secured by a set-screw or taper pin; being merely pressed on to the ends of the rocker arm, they are easily driven off, and probably are so arranged in order that they may automatically equalize any variation in valve-stem length that develops in work. The spindles of the valve rockers are carried directly in the camshaft housing, no gunmetal or other bushes being fitted. The inner arms of the rocker levers in contact with the cams are fitted with hardened-steel rollers.

The diameter of the valve stems is 11 mm., and each valve works in a phosphor bronze bush in the valve-stem guide, as already mentioned. All valves are set at an angle of 15 degrees to the cylinder center line, and single helical valve springs are fitted. They measure 39 mm. at the base of

the coil and taper to 31 mm. at the top. Conical split nuts, locked by the pressure of the valve spring on its washer, locate the springs on the valve stems; details of this arrangement are shown in the illustrations.

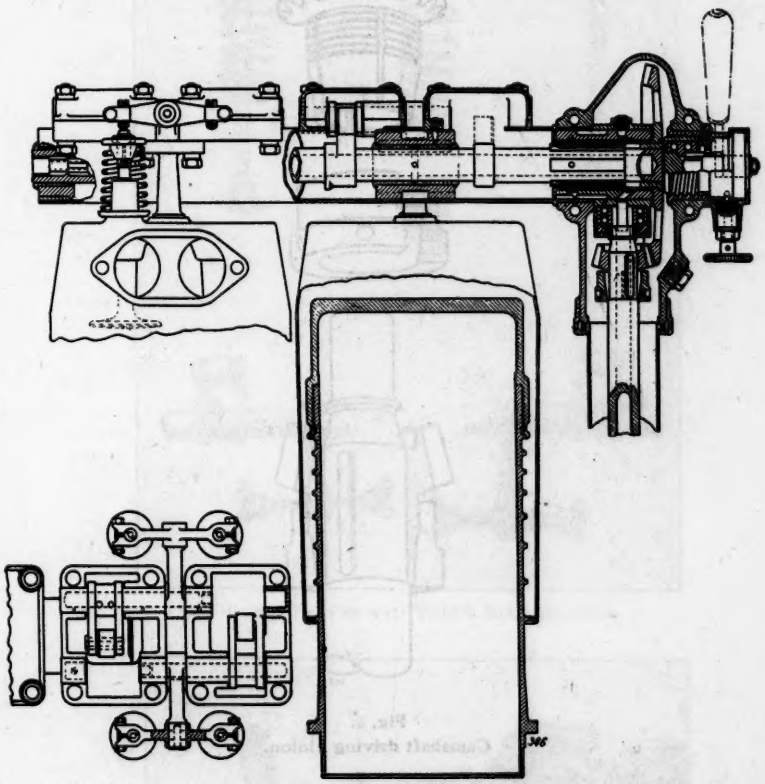


Fig. 5.
Arrangement of camshaft and half compression gear.

CAMSHAFT DRIVE AND HALF COMPRESSION GEAR.

The camshaft, as previously mentioned, is driven through a vertical shaft by the main distribution bevel pinion mounted on the end of the crankshaft by four splines. The method of attaching the bevel pinion to the top end of the vertical driving spindle is interesting. The fixing of this pinion is so designed that it may be adjusted for mesh in relation to the teeth of the camshaft-driven bevel wheel. Details of this gear and its fixing arrangement are shown in Fig. 6. The end of the shaft is 23 mm. in diameter and ground parallel; it is fitted with a key, and on this the bevel pinion is a push fit. A ground extension sleeve, tapered and screwed as shown in Fig. 6, is arranged on the bottom of the pinion, this sleeve

Fig. 6.
Camshaft driving pinion.

being split by four saw cuts, so that the application of the screwed collar closes in the slots and so locks the bevel on the shaft.

HALF COMPRESSION GEAR.

This is similar in design to that fitted on the 160 H.P. Mercédès engines, in which the camshaft is designed to slide longitudinally in its bearings and bring into operation a small cam situated on the mid-neutral axis of the exhaust cam proper. Sliding movement of the camshaft in its bearings is provided by arranging the camshaft driving bevel wheel upon castellations on the end of the camshaft. The camshaft driving wheel is mounted in a split gun-metal bearing in the rear end of the camshaft housing, and the sliding movement of the camshaft is effected by means of a hand lever set at right angles to the axis of the camshaft, and to which is attached a gun-metal collar screwed inside with a five-start square thread of 23 mm.

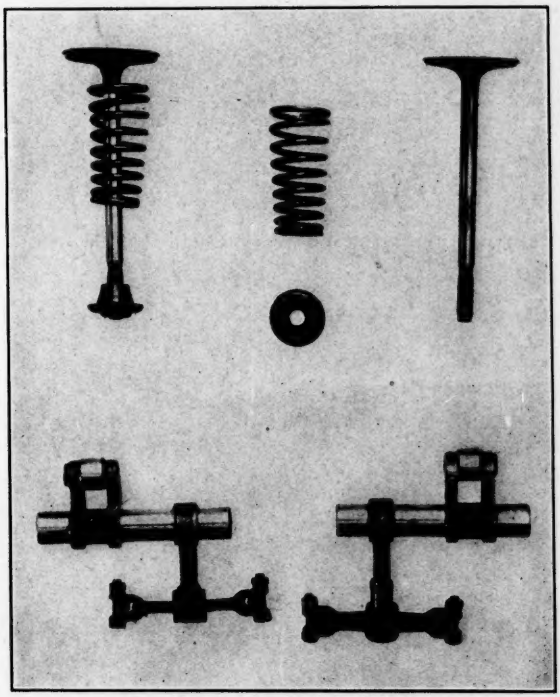


FIG. 4.—VALVES AND VALVE ROCKER ARMS.

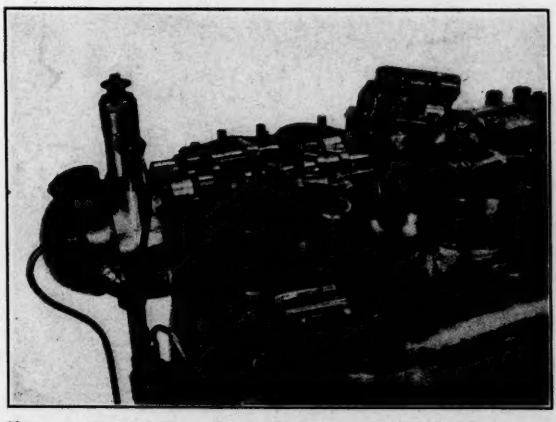


FIG. 5.—FRONT END OF CAMSHAFT AND VALVE ROCKERS.

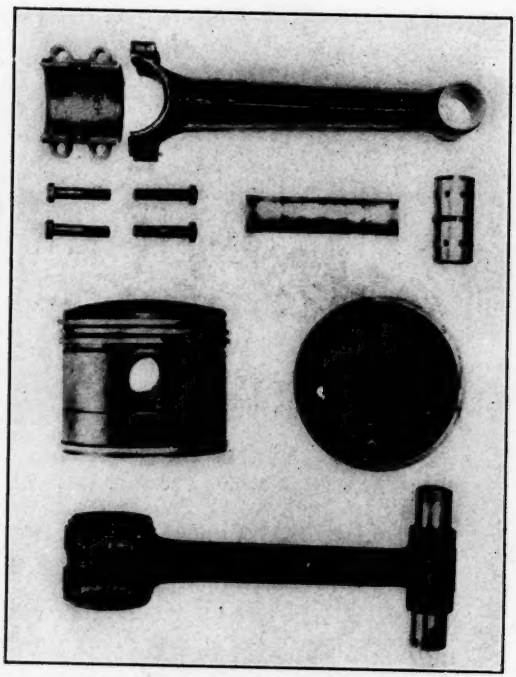


FIG. 8.—CONNECTING ROD, PISTON, GUDGEON PIN AND BUSH.

pitch. This collar, which is located in an aluminum housing, engages with a corresponding screw thread cut on the outer diameter of a steel sleeve. Inside the sleeve is a double ball thrust, which is fixed to the rear end of the camshaft, so that in operation the movement of the half compression lever about the axis of the camshaft rotates the quick-thread screw in the sleeve, thus pulling or pushing the end of the camshaft in its bearings through the castellations of the camshaft-driving bevel wheel. When the camshaft is drawn to its limit of movement towards the rear, the half-compression cam is drawn into line with the roller end of the exhaust-valve rocker arm, giving the requisite lift to the exhaust valves. The half-compression cam opens the exhaust valve near the end of the compression stroke.

The hardened-steel rollers on the cam rocker arms are chamfered off at 45 degrees on one side to allow of easy engagement of the half-compression cam, which is also chamfered at 45 degrees. The gear is clearly illustrated in the drawing of the rear end of the camshaft driving gear.

CONNECTING RODS.

These are of normal design and consist of H-section forgings. They measure 326 mm. between centers, and the total weight of the complete connecting rod, including the small end bush, is 7 pounds. The weight of the big end complete is 4 pounds 14 ounces, and the weight of the small end 2 pounds 2 ounces. The thickness of the center web of the H-section is 2.5 mm., while that of the two flanges is 3.5 mm. The diameter of the big end bearings is 64 mm. and the length 80 mm. These are split in the usual manner and held by four 12 mm. bolts.

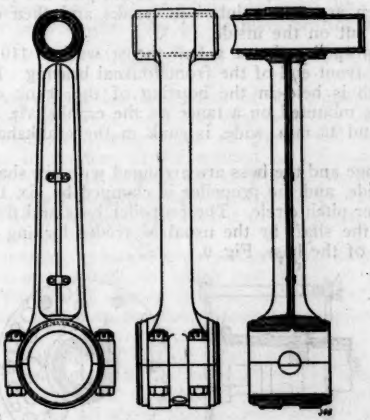


Fig. 7.
Connecting rod.

The distinctive feature of the connecting rods is the floating gudgeon-pin bush, the design of which is similar to that used on the 160 H.P. Mercédès. The bush is of hardened steel, ground 2.5 mm. in thickness, the outside diameter being 42 mm. and the inner diameter is 37 mm. The bush is a running fit in the small end and also on the gudgeon pin, and is pierced by fourteen 5 mm. holes in the same manner as that adopted upon the

floating sleeves of commercial vehicle road wheels. Oil is supplied by a 6-mm. pipe leading upwards from the big end bearings. The pipe is of soft mild steel, and attached to the web of the connecting rod by two riveted clips. Details of the connecting rods and floating gudgeon pin bush are shown in Figs. 7 and 8.

CRANKSHAFT.

The six-throw crankshaft, which is of very massive design, weighs 139.5 pounds, including the propeller boss. The cranks are set at 120 degrees, and the diameter of both the crank pins and journals is 64 mm. The length of the front journal bearing next to the propeller is 104 mm., and the length of all the other crankshaft bearings is 64 mm. The thickness and the width of the crank webs increases towards the front end of the crankshaft, so that the thickness of the front web is 29 mm., that of the second 28 mm., while the remainder are all 27 mm. in thickness.

The crankshaft is hollow, and the webs are drilled for the passage of oil from the journals to the crank pins for lubricating the big end bearings in the usual manner. The holes bored in the journals and crank pins for lightening purposes are of decreasing diameters, as follows:

Internal Diameter of Hollow Crankshaft Journals.

	1.	2.	3.	4.	5.	6.
Prop end						
Oil pump	32 mm.	33 mm.	35 mm.	38 mm.	40 mm.	41 mm.

Internal Diameter of Hollow Crank Pins.

	1.	2.	3.	4.	5.	6.
Prop end						
Oil pump	35 mm.	35 mm.	35 mm.	38 mm.	40 mm.	41 mm.

The ends of the hollow journals and crank pins are plugged with sheet-steel caps, which are driven into the cranks and then expanded into an annular groove cut on the inside.

To take the propeller thrust a ball thrust washer 110 mm. diameter is provided on the front end of the front journal bearing. It is mounted in a split cage, which is held in the housing of the crank case halves. The propeller boss is mounted on a taper on the crankshaft, and a single key, 120 mm. long and 13 mm. wide, is sunk in the crankshaft parallel to the taper.

The front flange and the boss are arranged with six shallow castellations, each 29 mm. wide, and the propeller is clamped by six 18 mm. bolts on a 230-mm. diameter pitch circle. The propeller boss and the front flange are both locked on the shaft by the usual Mercédès locking device, as shown in the drawings of the boss, Fig. 9.

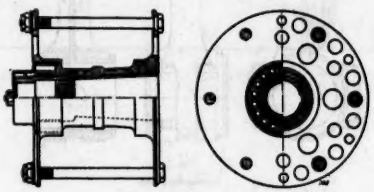


Fig. 9.
Arrangement of propeller boss.

CRANK CASE.

The very clean design of the crank case is evident from the illustrations, and, furthermore, it is of very light section considering the size of the engine, the average thickness of the walls of the casing being 6 mm. The

top and bottom halves are bolted together in the usual way by twenty-six 10-mm. hollow bolts through the flange on the center level of the crankshaft.

As already mentioned, the crankshaft main bearing caps are cast as part of the bottom half of the crank chamber, and the long 20-mm. bolts which secure the bearings caps at the bottom pass through the top half of the crank case and act as the holding-down bolts for the cylinders, which are held in position by triangular bridge clamps. The bottom ends of the cylinder barrels extend past the base flanges, registering into the top of the crank case, and the holding-down bolts clamp each cylinder-base flange at four points by the bridge pieces, or dogs, arranged between each pair of cylinders.

Below the crank chamber at the rear end is situated the main oil sump, which contains the oil pump, and in the front end of the base-chamber casting below the false bottom is an auxiliary-service oil sump, into which the return oil from the circulation system drains through a wire-gauze filter.

The central portion of the base-chamber casting, between the two sumps, forms an air chamber for the dual purpose of cooling the oil and warming the air supplied to the intake of the carburetor. Exterior air is led by two ventilating funnels on each side of the crank case into two oval ports in the base-chamber casting, and a number of fins are cast on the underside of the false bottom inside the base-chamber for cooling purposes.

The air intake to the carburetor is 4 inches in diameter, and it is cast in the left-hand side of the crank chamber and base, as shown in the general arrangement drawings of the engine. Cooling fins are also cast on the underside of the sump, and four breathers are fitted into the top half of the crank chamber between the webs of the main bearing housings.

LUBRICATION.

Although the general lubrication system of the engine is arranged on normal principles, it appears advisable to deal with the various details of the somewhat complicated plunger-type oil pump, and to describe their different functions in relation to the method of lubricating the various parts of the engine. The system may be sub-divided into three circuits:

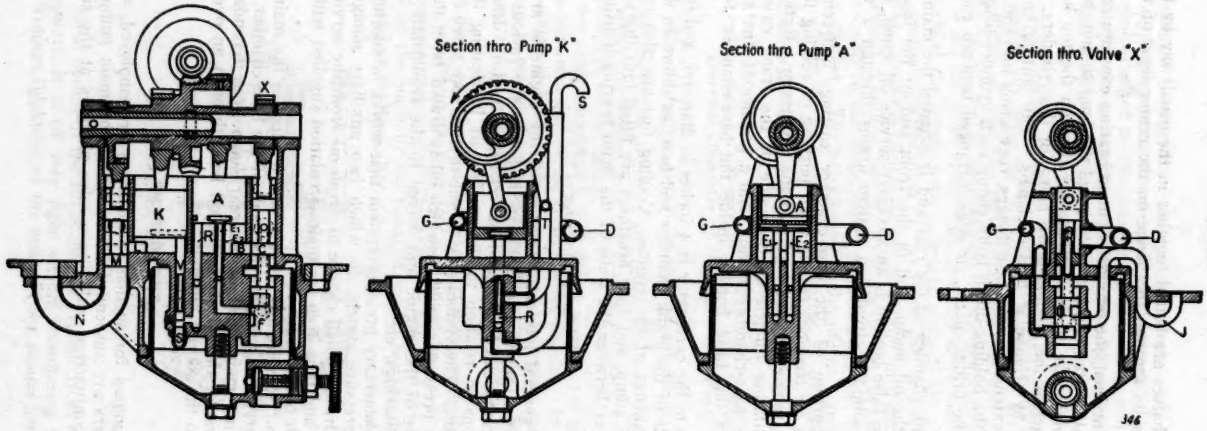
(*a*) The main pressure circuit, in which oil is drawn from the main oil sump at the rear of the engine and forced to the crankshaft connecting rods, camshaft bearings, etc.

(*b*) The supplementary pressure system that works in conjunction with the main high-pressure system, in which two auxiliary plungers of the main oil pump draw a small charge of fresh oil from the service oil tank at every stroke, and thus keep the main circulation supplied with a certain quantity of fresh oil.

(*c*) A suction "scavenging" circuit, which supplies the main oil sump from the auxiliary drain sump at the front of the base chamber, the working oil level being maintained in the rear sump by a secondary suction pump that draws off the oil above the oil level through an overflow pipe and returns it to the service oil tank.

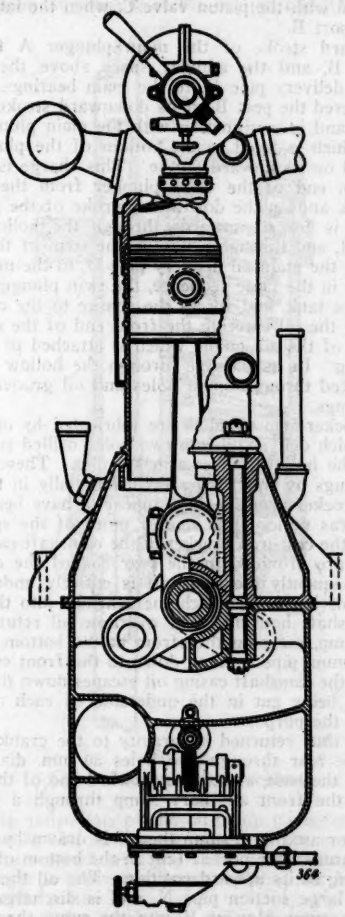
THE OIL PUMP.

Two main plungers for suction and delivery are employed, while three auxiliary plungers are suspended from one of the main pump plungers; these work in conjunction with piston-valve plungers at the sides of the main pump.



General arrangement of oil pump.

Each pump and valve plunger is operated by an eccentric, and the eccentric spindle is arranged transversely to the engine crankshaft; it is driven by worm gear from a layshaft at a 17 to 1 ratio to engine speed. The layshaft is driven by bevel gearing from the vertical driving-shaft of



Cross section through cylinder and crank case.

the water pump, this being in turn driven from the main bevel-gear pinion on the rear end of the crankshaft, as shown in the general arrangement drawing of the engine, Fig. 11.

The functioning of the oil pump will perhaps best be grasped by reference to the diagrammatic section (Fig. 12), and, considering first the

main pressure system of lubrication to the main bearings, its action is as follows: Oil is drawn from the main sump through the wire-gauze filter in the bottom of the oil-pump body by the suction of the main plunger pump A through the port B (which is cut in the bottom of the pump barrel), in conjunction with the piston valve C, when the latter is in the open position with the port B.

On the downward stroke of the main plunger A the oil is forced through the port B, and the annular space above the piston valve C, through the main delivery pipe D, to the main bearings, when the piston valve C has uncovered the port B on its downward stroke.

Simultaneously, and in conjunction with the main plunger A, the auxiliary plunger E, which is fitted to the bottom of the plunger A, draws a charge of fresh oil on its upward stroke. This charge is taken through a port in the bottom end of the valve plunger from the oil-service tank, through the pipe G, and on the downward stroke of the plunger pump E; the charge of oil is forced upwards through the hollow-valve plunger, through the port H, and through a hole in the stem of the plunger, which communicates with the main oil delivery pipe D, to the main bearings. At the same time, and in the same sequence, the twin plunger EI draws fresh oil from the service tank, and forces the charge to the camshaft bearings through the pipe J, the oil entering the front end of the camshaft through the hollow spindle of the air pump, which is attached to the front end of the camshaft casing. In its passage through the hollow camshaft the oil under pressure is fed through small holes and oil grooves cut in each of the camshaft bearings.

The camshaft rocker-arm spindles are lubricated by oil thrown off the revolving cams, which deliver oil into two holes drilled side by side in the upper portion of the hollow rocker-arm spindles. These holes communicate with the bearings by small holes drilled radially in the spindle. The lubrication of the rocker arm does not appear to have been very efficiently carried out, as it was noticed that one or more of the spindles had commenced to seize in the cast-iron bracket in the camshaft casing.

No outlet holes are provided in the rear end of the camshaft for the egress of oil; consequently the camshaft is entirely under a constant oil pressure, which finds its way through the bearings into the troughs in the bottom of the camshaft housing. The overflow oil returns to the crankcase via the air pump, escaping therefrom at the bottom of the air-pump crank case by a 12-mm. pipe running down to the front end of the engine. At the rear end of the camshaft casing oil escapes down the hollow vertical shaft, oil channels being cut in the underside of each of the camshaft-bearing bushes for the purpose.

The oil that has thus returned by gravity to the crank case is led into the oil sump at the rear through two holes 30 mm. diameter, drilled in the false bottom of the base, and at the forward end of the crank chamber the oil flows into the front auxiliary sump through a wire-gauze filter already referred to.

From the front, or auxiliary, sump the oil is drawn by the "scavenger" pump K, Fig. 12, through the port L (cut in the bottom of the barrel), the piston valve M being in its upward position. The oil therefore enters the pump through the large suction pipe N, and is discharged on the downward stroke of the pump plunger K into the sump through the port P, drilled in the side of the piston-valve barrel.

To maintain the correct working oil level in the sump, a third plunger R, also attached to the bottom of the main plunger pump A, draws off, through the bent-over suction pipe S, any surplus oil above the working level in the sump, and returns it to the oil-service tank through the pipe T, on the downward stroke of the plunger R, this plunger working in conjunction with the piston valve V, which is flexibly attached to the bottom

of the suction pump K, in a slot cut in the face of the pump. In the main pressure delivery pipe to the crankshaft bearings, and directly above the oil pump, the pulsation-damper valve is located. The details of this valve are shown in Fig. 10, while its exact position may be seen by reference to the general arrangement drawing of the engine, Fig. 11.

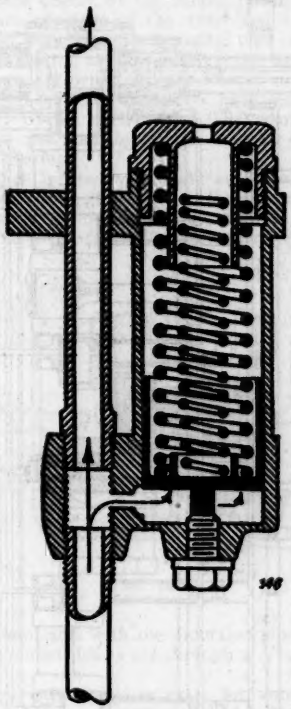


Fig. 10.
Oil pulsation damper valve.

In operation the pulsations of the oil-pump plungers are damped out in the delivery pipe by the cushioning effect of the spring-loaded plunger, which communicates direct with the sump on its underside. The bottom face of the plunger is prevented from reaching the bottom of the barrel by a small set screw in the base.

The lubrication of the crankshaft and connecting-rod bearings is on standard lines, and has already been referred to; details of this are shown in the general arrangement of the engine.

CARBURETOR AND INDUCTION SYSTEM.

Notwithstanding the large size of the engine, only one carburetor is employed, and, as will be seen from the illustrations, this is situated low down at the rear end of the crank case. Below the float chamber a petrol-

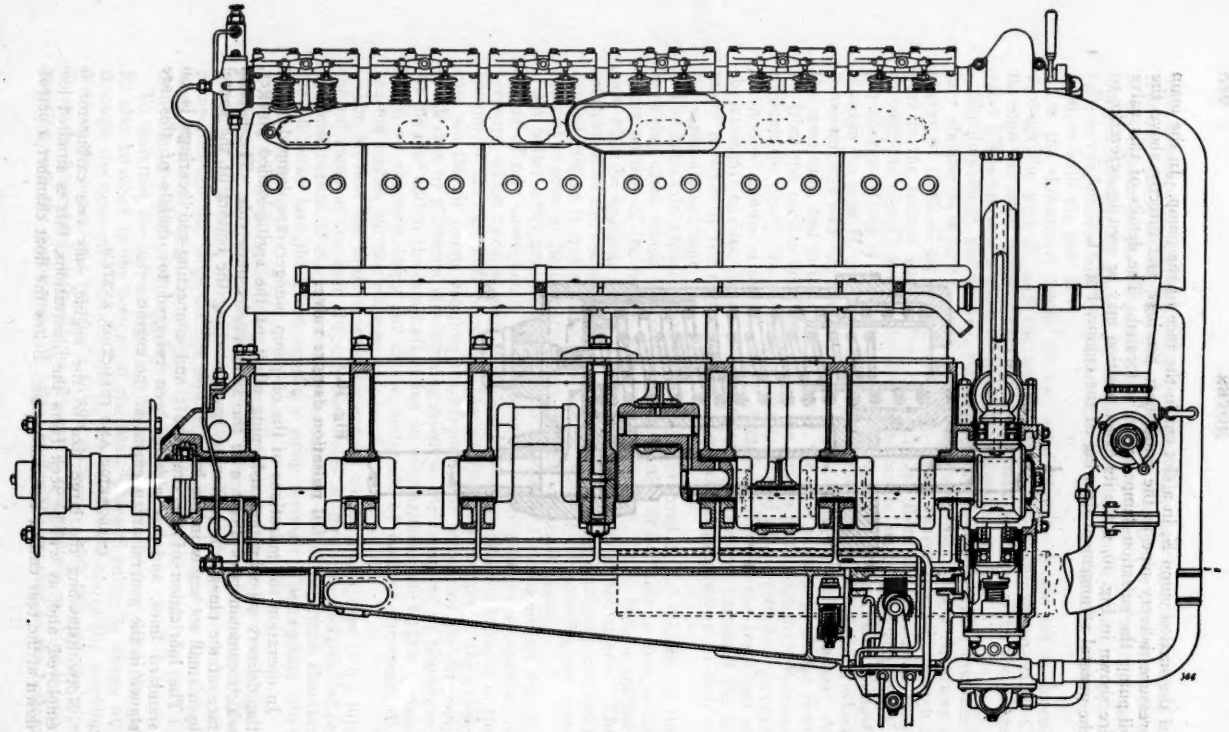


Fig. 11.

GENERAL ARRANGEMENT OF 260 H.P. MERCÉDÈS AIRCRAFT ENGINE. 160 MM. BORE × 160 MM. STROKE.

filter chamber is attached, the petrol entering through a gauze cylindrical filter tube that is screwed into the top of the filter chamber, as shown in Fig. 13. Petrol enters the bottom of the float chamber, which is of the ordinary balanced needle valve type.

The main jet, which is a plain tube having an orifice of 2.3 mm. diameter, is situated in the center of the intake inside the choke tube. The choke tube is 32 mm. diameter at the waist and 54 mm. diameter at the largest diameter top and bottom. A barrel-type of throttle valve is arranged, 80 mm. in diameter, and mounted on the ball bearings at each end. The races are 35 mm. diameter, and are located in recesses turned in the end covers of the gun-metal throttle-valve liner, which is pressed into the cast-aluminum body of the carburetor. The semi-elliptical ports in the throttle valve are 80 mm. long by 50 mm. wide at the bottom side and 80 mm. by 55 mm. at the top.

The pilot jet, which is the same length as the main jet, is .9 mm. bore at the orifice, and is situated at the side of the intake passage. It communicates with an annular groove machined around the outer end of the throttle-valve liner in the body of the carburetor. From this annular chan-

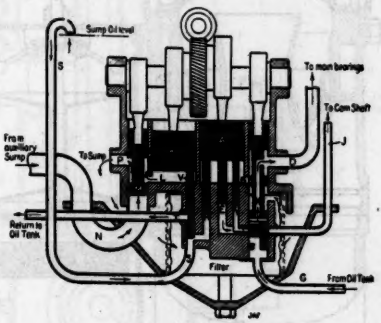


Fig 12.
Diagram of oil pump.

nel a passage communicates with the induction pipe just above the throttle valve, which when closed draws air through a V slot cut in the barrel of the throttle.

A conical suction valve supplies extra air automatically at all speeds through eight holes, 14 mm. diameter, drilled in the annular seating which surrounds the base of the choke tube. The air to the carburetor, as already mentioned, is taken through the interior of the base chamber; the diameter of the air intake passage is 100 mm., and it extends inside the crank case to the center of the engine.

The throttle valve is water jacketed by pipes leading from the main water-circulation pipe to water spaces cast in the body of the carburetor around the throttle valve.

INDUCTION PIPE.

The diameter of the induction pipe at the joint where it meets carburetor is 70 mm., this diameter increasing to 100 mm. towards the engine, where it takes a right-angled bend. It is lagged with asbestos cord up to this point.

The design of the top portion of the induction pipe is interesting, the gas being led to the center of the engine by the 100-mm. pipe, whence it enters two branches 70 mm. diameter, through a port 100 mm. diameter.

The two branches lead outwards to the inlet ports, three, of course, on either side of the main entry. One-half of the branch is enclosed by the main induction pipe. The cylinder ports are 85 mm. long and 50 mm. wide.

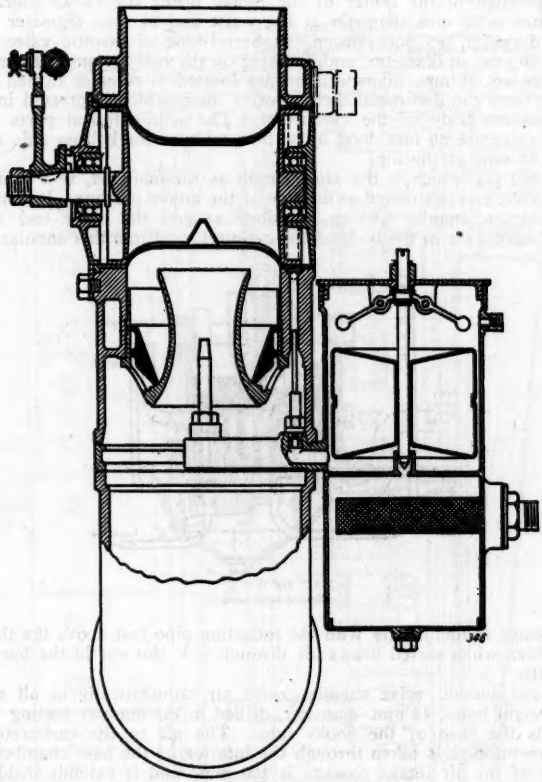


Fig 13
Arrangement of carburetter.

IGNITION.

Two Bosch magnetos, type Z.H.6, are mounted transversely to the crankshaft upon a bracket at the rear end of the crank case, one being a starter magneto used in conjunction with a hand-starter dynamo. The magnetos are driven by bevel gears directly off the vertical camshaft driving spindle, the direction of rotation of both being anti-clockwise. The arrangement of the magneto driving gear is shown in Fig. 14.

All twelve sparking plugs are fitted on the inlet side of the engine, and are situated directly below the inlet valves, the high-tension wire being

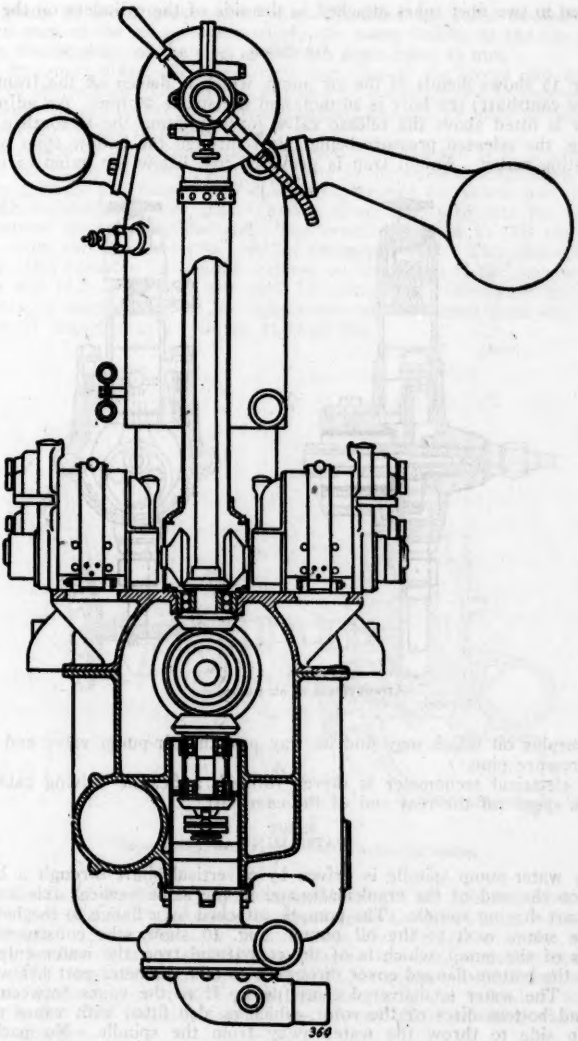


Fig. 14.
Arrangement of camshaft vertical drive.

carried in two fiber tubes attached to the side of the cylinders on the inlet side.

AIR PUMP.

Fig. 15 shows details of the air pump, which is driven off the front end of the camshaft; the bore is 26 mm. and the stroke 27 mm. An adjusting screw is fitted above the release valve for regulating the strength of the spring, the released pressure being taken through the hollow stem of the adjusting screw. An oil trap is provided just below the pump to retain

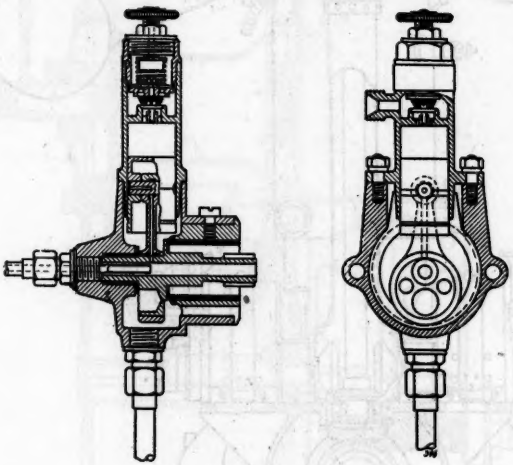


Fig. 15.
Arrangement of air pump.

any surplus oil which may find its way past the air-pump valve and into the pressure pipe.

An electrical tachometer is driven through a flexible driving cable at engine speed off the rear end of the camshaft.

WATER PUMP.

The water-pump spindle is driven by a vertical shaft through a bevel gear on the end of the crankshaft, and in the same vertical axis as the camshaft driving spindle. The pump is attached by a flange to the bottom of the sump, next to the oil pump. Fig. 16 shows the constructional details of the pump, which is of the centrifugal type, the water entering below the bottom flanged cover through a 45-mm. diameter port below the rotor. The water is delivered centrifugally from the vanes between the top and bottom discs of the rotor, which is also fitted with vanes upon its top side to throw the water away from the spindle. No packing glands are provided for the driving spindle, but a hardened-steel washer is set into a recess machined in the upper face of the rotor, and this is kept in uniform contact with the face of the phosphor bronze spindle bush by the action of a light spring that is fitted under a ball-thrust race at the driving end of the spindle. The diameter of the outlet passages from the

water pump to the cylinders is 45 mm. Double-inlet water connections between each of the six cylinders supply the water jackets at the top and bottom, the diameter of the steel circulation pipes being 40 mm.

The foregoing notes will give a very fair idea of the layout and detail design of the engine. The figures, analyses of materials, are in process of preparation by the Naval and Military authorities, and we hope to include these further details in a near issue.

With regard to the actual detail construction of the 260 H.P. Mercédès engine, practically the only point of novelty is the scheme adopted on the built-up cylinder and head. It is difficult to arrange for valves per cylinder with satisfactory water spaces, and it should be noted that the valve arrangement on the Mercédès engine is practically good in this respect, ample room for free water circulation being provided. The importance of this point becomes increasingly evident on an engine of the dimensions of the 260 H.P. Mercédès, not only on account of immediate cooling problems in running, but in the subsequent effect of such heat and the dangers of distortion in a cylinder of large size.

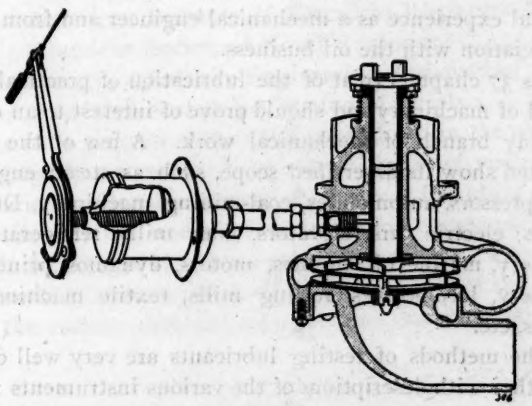


Fig. 16.
Water pump, with greaser and its actuating ratchet.

The details of the built-up cylinder and head are all different from any other engine that has at present emanated from Germany, and, occurring in an engine of the Mercédès manufacture, this departure is the more marked, because it is an entirely new scheme for this particular firm to adopt. The trend of design on the 260 H.P. Mercédès cylinders and heads is most markedly towards Maybach practice, and the inference to be drawn is that this particular system has proved itself eminently satisfactory in practice.

The 260 H.P. Mercédès, it must be admitted, is a workmanlike and nicely completed job, homogeneous in its general layout, and with many minor pleasing points for the scrutinizing designer.—"Aerial Age Weekly."

BOOK REVIEW.

LUBRICATING ENGINEER'S HANDBOOK. By JOHN ROME BATTLE, B. S. in M. E. Published by J. B. LIPPINCOTT CO., Philadelphia, Penn., 1916. Cloth; 6 × 9 in.; 333 pp.; 114 illustrations; 25 tables. Price, $4.00.

This is a reference book of data and tables and general information for designers of machinery, operating engineers, owners and general managers, purchasing agents, lubricating oil salesmen and manufacturers of lubricants, written from actual experience as a mechanical engineer and from years of association with the oil business.

Its 37 chapters treat of the lubrication of practically every kind of machinery and should prove of interest to an engineer in any branch of mechanical work. A few of the features treated show its diversified scope, such as steam engines, air compressors, automobiles, coal-mining machinery, Diesel engines, electric cars, elevators, flour mills, refrigerating machinery, marine oil engines, motors, dynamos, printing machinery, locomotives, rolling mills, textile machinery, turbines, etc.

The methods of testing lubricants are very well outlined, together with descriptions of the various instruments required for testing.

The book contains a great deal of practical data which would make it of value to the operating engineer. It might have a wider field of usefulness were more technical data contained in the volume. A parallel volume presenting the same matter in a more technical manner would supplement this book very nicely.

ASSOCIATION NOTES.

There recently has been organized in Honolulu, Hawaii, The Engineering Societies Club of Hawaii, the present prerequisite to eligibility for membership being membership of whatever grade in one of the following:

American Society of Civil Engineers,
American Society of Mechanical Engineers,
American Institute of Mining Engineers,
American Institute of Electrical Engineers,
American Society of Naval Engineers,
American Railway Engineering Association,
American Society for Testing Materials.

The main purpose of the Club, as set forth in the notice received by the Secretary, is the entertainment of visitors from the various national organizations while in Honolulu and the Hawaiian Islands, and a cordial invitation is extended to any of the members of the above mentioned societies who may go to Hawaii to announce himself to the Secretary of the Club so that the various members may get into communication with him.

The regular meeting of the Society for the nomination of officers for the year 1918 was held at the Navy Department, Washington, D. C., on Tuesday, October 2, 1917. The following were nominated:

For *President:*

Captain H. P. Norton, U. S. N.

For *Secretary-Treasurer:*

Lieutenant Commander F. W. Sterling, U. S. N. (Retired).

For *Member of Council:*

Captain T. W. Kinkaid, U. S. N.
Captain W. Strother Smith, U. S. N.
Naval Constructor Robert Stocker, U. S. N.
Commander W. W. White, U. S. N. (Retired).
Lieutenant Commander J. O. Fisher, U. S. N.
Lieutenant Commander S. M. Robinson, U. S. N.
Lieutenant Commander H. T. Dyer, U. S. N.
Engineer Constructor John Q. Walton, U. S. Coast Guard.

The meeting for counting ballots cast will be held at the Navy Department, Washington, D. C., on Saturday, December 29, 1917.

It was the sense of the meeting that, owing to the unsettled conditions obtaining, it would be inadvisable to hold a banquet of the Society during the year 1918.

The following members and associates have joined the Society since the publication of the last JOURNAL:

MEMBERS.

Piersol, W. Burton (ex-Naval officer) 325 South 44th St., Philadelphia, Pa.

ASSOCIATES.

Calhoun, Guy K., Professor of Mathematics, U. S. N.
Charles, W. J., Ensign, N. N. V., Armed Draft Office, Park Barracks, Navy Yard, New York City.
Conradson, Pontus, Chief Chemist, Galena-Signal Oil Co., Franklin, Pa.
Petersen, Carl Edward, 5616 Fourth Avenue, Brooklyn, N. Y.
Smith, Charles Howard, Consulting Engineer, 24 Broad St., New York City.
Wherrett, W. H., Department M. E. and N. C., Naval Academy, Annapolis, Md.

ADDRESSES.

In view of prevailing conditions the Secretary earnestly requests that each member keep him fully informed of the address to which the JOURNAL is to be delivered.

A set of the JOURNAL, Vol. I to XXVIII, inclusive, is offered for sale. Vols. I to XXV, inclusive, are bound. Price for bound volumes $6.50 each; for unbound volumes $5.00 each. All in good condition. Inquiries should be addressed "J., care of American Society of Naval Engineers, Navy Department, Washington, D. C."